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~~PROCESS AND DEVICE FOR CONTROLLING THE THERMAL AND ELECTRICAL OUTPUT OF INTEGRATED MICRO COMBINED HEAT AND POWER GENERATION SYSTEMS~~

INTEGRATED MICRO COMBINED HEAT AND POWER SYSTEM

CROSS REFERENCE TO RELATED APPLICATIONS

[0001] This application is a divisional of co-pending and now-allowed Application Serial Number 10/146,006, filed May 15, 2002, which is a continuation of U.S. Patent Application Serial No. 09/998,705 filed November 30, 2001 (now US Patent 6,598,397), which claims the benefit of U.S. Provisional Application No. 60/311,514 filed August 10, 2001.

BACKGROUND OF THE INVENTION

~~The present invention relates to cogeneration systems and, more particularly, to microprocessor-based control sub-systems for controlling the thermal and electrical output of integrated micro-combined heat and power generation systems used to supply domestic electrical power, domestic space heating (SH) water, and domestic hot water (DHW).~~

[0002] The present invention generally relates to a cogeneration system for the supply of electrical power, space heating (SH) water and domestic hot water (DHW), and more particularly to a small scale Rankine-type cogeneration system that utilizes a scroll expander and an organic working fluid.

[0003] The concept of cogeneration, or combined heat and power (CHP), has been known for some time as a way to improve overall efficiency in energy production systems. With a typical CHP system, heat (usually in the form of hot air or water) and electricity are the two forms of energy that are generated. In such a system, the heat produced from a combustion process can drive an electric generator, as well as heat up water, often turning it into steam for dwelling or process heat. Most present-day CHP systems tend to be rather large, producing heat and power

for either a vast number of consumers or large industrial concerns. Traditionally, the economies of scale have prevented such an approach from being extrapolated down to a single or discreet number of users. However, increases in fuel costs have diminished the benefits of centrally-generated power. Accordingly, there is a potentially great market where large numbers of relatively autonomous, distributed producers of heat and electricity can be utilized. For example, in older, existing heat transport infrastructure, where the presence of fluid-carrying pipes is pervasive, the inclusion of a system that can provide CHP would be especially promising, as no disturbance of the adjacent building structure to insert new piping is required. Similarly, a CHP system's inherent multifunction capability can reduce structural redundancy.

[0004] The market for localized heat generation capability in Europe and the United Kingdom (UK), as well as certain parts of the United States, dictates that a single unit for single-family residential and small commercial sites provide heat for both SH (such as a hydronic system with radiator), and DHW (such as a shower head or faucet in a sink or bathtub), via demand or instantaneous system. Existing combination units are sometimes used, where heat for DHW is accumulated in a combination storage tank and boiler coil. In one configuration, SH water circulates through the boiler coil, which acts as the heating element for the water in the storage tank. By way of example, since the storage capacity required for instantaneous DHW supplying one to two showers in a single family residence (such as a detached house or a large apartment) is approximately 120 to 180 liters (roughly 30 to 50 gallons), it follows that the size of the storage tank needs to be fairly large, sometimes prohibitively so to satisfy thermal requirements of up to 25 kilowatts thermal (kWt) for stored hot water to meet such a peak shower demand. However, in newer and smaller homes there is often inadequate room to accommodate such storage tank volume. In addition to the need for instantaneous DHW capacity of up to 25 kWt, up to 10 kWt for SH is seasonally needed to heat an average-sized dwelling.

[0005] Furthermore, even in systems that employ SH and DHW in a single heating system to consolidate spacing, no provision for CHP is included. In the example given above, it is likely that the electrical requirements concomitant with the use of 35 kWt will be between 3 and 5

kilowatts electric (kWe). The traditional approach to providing both forms of power, as previously discussed, was to have a large central electricity generating station provide electricity on a common grid to thousands or even millions of users, with heat and hot water production capacity provided at or near the end-user on an individual or small group basis. Thus, with the traditional approach, the consumer has not only little control over the cost of power generation, as such cost is subject to prevailing rates and demand from other consumers, but also pays more due to the inherent inefficiency of a system that does not exploit the synergism of using otherwise waste heat to provide either additional electric generation or heating capacity.

[0006] Large-scale (in the megawatt (MW) range and up) cogeneration systems, while helpful in reducing the aforementioned inefficiencies of centrally-based power generation facilities, are not well-suited to providing small-scale (below a few hundred kW) heat and power, especially in the small-scale range of a few kWe and below (micro-based systems) to a few dozen kWe (mini-based systems). Much of this is due to the inability of the large prime mover systems to scale down, as reasonable electrical efficiency is often only achieved with varying load-responsive systems, tighter dimensional tolerances of key components, and attendant high capital cost. Representative of this class are gas turbines, which are expensive to build for small-scale applications and which sacrifice efficiency when operating over varying electrical load requirements. Efficiency-enhancing devices, such as recuperators, tend to reduce heat available to the DHW or SH loops, thus limiting their use in high heat-to-power ratio (hereinafter Q/P) applications.

[0007] A subclass of the gas turbine-based prime mover is the microturbine, which includes a high-speed generator coupled to power electronics. This microturbine could be a feasible approach to small-scale cogeneration systems. Other shortcomings associated with large-scale CHP systems stem from life-limited configurations that have high maintenance costs. This class includes prime movers incorporating conventional internal combustion engines, where noise, exhaust emissions, lubricating oil and spark plug changes and related maintenance and packaging requirements render use within a residential or light commercial dwelling

objectionable. This class of prime mover also does not reject a sufficient amount of heat for situations requiring a high Q/P, such as may be expected to be encountered in a single family dwelling. Other prime mover configurations, such as steam turbines, while generally conducive to high Q/P, are even less adapted to fluctuating electrical requirements than gas turbines. In addition, the steam-based approach typically involves slow system start-up, and high initial system cost, both militating against small-scale applications.

[0008] In view of the limitations of the existing art, the inventors of the present invention have discovered that what is needed is an autonomous system that integrates electric and heat production into an affordable, compact, efficient and distributed power generator, and the ability to control the thermal and electrical output of such a system.

SUMMARY OF THE INVENTION

~~The above mentioned needs are meet by providing microprocessor based control sub-systems which control the thermal and electrical output of integrated micro combined heat and power generation (M-CHP) systems used to supply domestic electrical power, domestic space heating (SH) water, and domestic hot water (DHW). The M-CHP system uses a microprocessor controller to control the internal operating conditions, such as pump speeds, gas flow rate, and evaporator outlet temperature. Controlling these parameters enables setting the capacity of the system at any instant in time, thereby permitting load following, using a variable capacity operation. The controller also monitors a number of additional safety controls and system protection devices, such as relays/contactor of the alternator to grid, and electrical trips to the feed pump, the oil pump, the hydronic pump, the blower, the gas valve, the expander bypass valves, and other electrically powered devices in the system.~~

~~Overall, system capacity control emphasizes simplicity, reliability, and low cost. For the most part, the most basic control system is a single flow rate system with on/off control from a space thermostat, wherein the instantaneous capacity of a hydronic heating system of M-CHP system is fixed. In a more preferred system, working fluid flow rate and system thermal capacity are~~

~~ultimately determined by the processor reading the outdoor ambient temperature. The outdoor ambient temperature is used to determine a set point for the supply (hot water) temperature in the hydronic heating system. The controller uses a look-up table or algorithm to establish the set point according to a linear scale, such as varying the set point linearly from 25ºC at an outdoor temperature of 20ºC to 75ºC at an outdoor temperature of - 20ºC. The controller uses this set point to operate the M-CHP in a variable capacity mode by modulating the pump flow rate to maintain the actual hydronic supply temperature at the desired set point.~~

~~Additionally, the controller coordinates the burner fuel flow rate (heat input rate) with the feed pump flow rate to provide optimal thermodynamic performance by preventing liquid from entering the expander. This is accomplished by maintaining a fixed evaporator exit temperature, such as, for example, 310ºF. Furthermore, the controller controls the hydronic pump of the heating system at a speed that keeps the pressure difference between the supply and return headers of the condenser at a preselected value for optimal thermodynamic performance of the hydronic heating system.~~

~~In normal operation the M-CHP system capacity will vary as needed to maintain the supply temperature at its set point, wherein the burner operates to provide the needed heat input. System mass flow varies essentially with pump speed, but since the expander operates at nearly a fixed speed, the volume flow of high pressure vapor is essentially constant. Thus, the evaporator pressure varies with load, allowing the vapor density at the evaporator exit to vary to maintain a constant volume flow at each mass flow setting. Preferably, a variable frequency drive is used for the feed pump motor. However, a motor with discrete speed steps may also be used, as well as any other variable speed or flow approach. In normal operation the system operates at higher capacity to attain the higher hydronic supply temperatures required by colder weather. With the higher hydronic supply temperature, the capacity of the indoor radiators/convectors increases to handle the added heat flow. After M-CHP shut down, the hydronic pump may run on to deliver residual heat in the hydronic system to the radiators and maintain the flow of valid control data to the controller from the hydronic system.~~

~~In one embodiment, coordinating the flow rates consists of open loop speed control of the feed pump with an induction motor. Proportional integral differential (PID) control of the boiler gas flow to maintain an outlet superheat between about 20°F and about 30°F using a closed loop control system may also be provided. A liquid condensate reservoir ahead of the feed pump and the liquid level is maintained above a predetermined level to ensure adequate net positive suction head (NPSH) to the pump. Preferably, no active control is necessary as a sufficient minimum level can be maintained under all operating conditions by filling the loop with a minimum refrigerant quantity for most operating conditions. However, if for certain operating conditions, the liquid level at the pump inlet cannot be maintained within a desired range, then an active means of maintaining liquid level control may be used. For example, level sensing at the liquid reservoir along with some means of speed control of the feed pump motor may be used.~~
~~High and low side pressures float depending mainly on boiler heat input and hydronic loop return temperature to the condenser. Additionally, no active control of the power module speed is necessary. Connection of the generator to the grid loads the alternator sufficiently to limit the speed to about 3150 RPM (50 hz application). However, should the controller sense a power module overspeed (e.g., loss of grid connection, by sensing, for example, the rate of change of frequency), it will quickly command the opening of a bypass loop around the expander. Preferably, the sensor to detect expander overspeed is built into the power module.~~

~~Typically, it is unnecessary for the controller to control the generator output as long as the generator output is less than the dwelling load. However, the controller can be programmed to monitor and deal with the situation when that generator output does exceeds dwelling load. In particular, the controller can be programmed to switch the extra power on the public grid or use the extra power for a resistance heater to preheat boiler feed.~~

[0009] These needs are met by the present invention, where a new micro-CHP system is described. Exemplary micro-cogeneration systems are disclosed by co-owned U.S. Patent Application No. 09/998,705 filed November 30, 2001, the entire disclosure of which is herein

incorporated fully by reference. In micro-CHP, a compact prime mover can provide both electric output, such as from a generator coupled to a heat source, as well as heat output to provide warm air and hot water to dwellings. What distinguishes micro-CHP from traditional CHP is size: in the micro-CHP, electric output is fairly small, in the low kW_e or even sub-kW_e range. The system of the present invention can provide rapid response to DHW requirements, as the size of tanks needed to store water are greatly reduced, or possibly even eliminated. The size of the micro-CHP system described herein can be adapted to particular user needs; for example, a system for a single-family dwelling could be sized to produce approximately 3 to 5 kW_e, 10 kW_t SH and 25 kW_t DHW. For small commercial applications or multi-dwelling (such as a group of apartment units) use, the system could be scaled upwards accordingly. The heat to power ratio, Q/P, is an important parameter in configuring the system. For most residential and small commercial applications, a Q/P in the range of 7:1 to 11:1 is preferable, as ratios much lower than that could result in wasted electrical generation, and ratios much higher than that are not practical for all but the coldest climates (where the need for heating is more constant than seasonal). Since the production of electricity (through, for example, a generator or fuel cell) is a byproduct of the prime mover heat generation process, no additional carbon dioxide and related atmospheric pollutants are generated, thus making the system of the present invention amenable to stricter emission control requirements.

[0010] According to a first aspect of the present invention, a cogeneration system configured to operate with an organic working fluid is disclosed. The system includes a heat source, a first circuit configured to transport the organic working fluid, and a generator operatively coupled to a scroll expander to produce electricity. The first circuit includes a scroll expander configured to receive the organic working fluid, a condenser in fluid communication with the scroll expander, and a pump configured to circulate the organic working fluid. The first circuit is in thermal communication with the heat source such that heat transferred therefrom converts the organic working fluid to a superheated vapor. The use of organic working fluid, rather than a more readily-available fluid (such as water) is important where shipping and even some end uses could subject portions of the system to freezing (below 32° Fahrenheit). With a water-filled system,

damage and inoperability could ensue after prolonged exposure to sub-freezing temperatures. In addition, by using an organic working fluid rather than water, corrosion issues germane to water in the presence of oxygen, and expander sizing or staging issues associated with low vapor density fluids, are avoided. The organic working fluid is preferably either a halocarbon refrigerant or a naturally-occurring hydrocarbon. Examples of the former include R-245fa, while examples of the latter include some of the alkanes, such as isopentane. Other known working fluids and refrigerants, despite exhibiting attractive thermodynamic properties, are precluded for other reasons. For example, R-11 is one of a class of refrigerants now banned in most of the world for environmental reasons. Similarly, R-123, much less environmentally objectionable (for now) than R-11, is the subject of decomposition concerns under certain micro-CHP operating conditions. The need to operate the condenser at a high enough temperature to allow useful hydronic space heating and the need to have a substantial vapor expansion ratio (of 5 to 7 or 8) limits the number of fluids with useful properties. In addition, the need to have a substantial vapor density at the expander inlet has a direct impact on the fluid choice and the diameter of the scrolls, both of which impact scroll cost. With many fluids, the condensing temperature and need for significant expansion result in very high scroll inlet pressures (increasing pumping power) or super critical conditions at the inlet, resulting in difficulties in evaporator design operation and control. These same conditions are of concern when one considers other natural (hydrocarbon) fluids. For example, while pentane, butane, and propane were all considered as potential working fluids, the inventors determined that, of the naturally-occurring hydrocarbons, isopentane offers superior fluid properties for micro-CHP applications.

[0011] According to another aspect of the present invention, a cogeneration system is disclosed. The cogeneration system includes an organic working fluid, a heat source capable of superheating the organic working fluid, a first circuit to transport the organic working fluid, and a generator to produce electricity. At least a portion of the first circuit, which includes a scroll expander, a condenser and a pump, is in thermal communication with the heat source. The pump circulates the organic working fluid through the first circuit. Preferably, the heat source is a burner in thermal communication with an evaporator such that heat provided by the burner

causes the organic working fluid that flows through the evaporator to become superheated. In the present context, the term "thermal communication" is meant to broadly cover all instances of thermal interchange brought about as a result of coupling between system components, whereas the more narrow "heat exchange communication" (discussed below) is meant to cover the more specific relationship between direct, adjacent heat exchange components designed specifically for that purpose. By the nature of the organic working fluid, it remains in a superheated state from prior to entering the scroll expander to after it exits the same. The high vapor density and heat transfer properties of the superheated organic working fluid ensure that maximum heat and power can be extracted from the fluid without having to resort to a large expander.

[0012] The cogeneration system can be configured such that the organic working fluid is directly-fired or indirectly-fired. In the former configuration, the relationship between the burner and the organic working fluid-carrying evaporator is such that the flame from the combustion process in the burner directly impinges on either the conduit carrying the fluid or a container (alternately referred to as a combustion chamber) that houses at least a part of the organic working fluid-carrying conduit such that the part of the conduit where the organic working fluid becomes superheated is considered the evaporator. In the latter configuration, the flame from the combustion process in the burner gives up a portion of its heat to conduit making up a secondary circuit, which in turn conveys a heat exchange fluid to an interloop heat exchanger. The heat exchange fluid could be water, a mixture of water and a freeze-inhibiting additive (such as propylene glycol), or an organic, such as that of the organic working fluid of the first circuit. The first loop of the interloop heat exchanger is fluidly connected to the organic working fluid-conveying first circuit, while the second loop is fluidly connected to the heat exchange fluid-conveying second circuit. Preferably, the interloop heat exchanger is situated between the pump and the scroll expander of the first circuit so that it acts as an evaporator for the organic working fluid. The latter configuration may also include a space heating loop preheat device that is in heat exchange communication with the condenser second loop such that a portion of the heat still present in the heat exchange fluid after giving up a portion of its heat to the organic working fluid in the interloop heat exchanger can be used to preheat fluid in an external SH loop.

[0013] Also, as with the former configuration, the burner can be disposed within a container. In both configurations, the container may include an exhaust duct to carry away combustion products (primarily exhaust gas), an exhaust fan to further facilitate such product removal, as well as an exhaust gas heat exchanger disposed adjacent (preferably within) the exhaust duct so that residual heat present in the exhaust gas can be used for supplemental heating in other parts of the cogeneration system. The exhaust gas heat exchanger can further include an exhaust gas recirculation device to further improve heat transfer from the exhaust gas. In the former configuration, the heat picked up by the exhaust gas heat exchanger can be routed to various places within either the first circuit or the space heating loop to provide additional preheat of the organic working fluid or space heating fluid, respectively. In addition, either configuration may be adapted to exchange heat with an external DHW loop. The heat exchange may further take place in a heat exchanger configured similar to the condenser, such that two individual loops are placed adjacent one another to facilitate the transfer of heat between the respective fluids flowing therethrough, or in a storage tank (such as a hot water storage tank) such that the fluid stored therein (preferably water) is kept at an elevated temperature to have a readily-available supply of hot tap, bath and shower water. In the case of a storage tank-based approach, additional heating of the liquid in the tank can occur by a heating element that receives its power from the generator. Where no tank is present, the heat to the DHW loop can be taken from a connection to the first circuit condenser (in the directly-fired configuration) or the heat exchange fluid flowing through the second circuit (in the indirectly-fired configuration). Furthermore, in either of the directly-fired or indirectly-fired configurations, if it is desired to preserve the ability to provide DHW while maintaining an overall simplistic, low-cost system, an oversized or multiple-staged burner may be used. This prompt heating can reduce the size of or even obviate the need for a large storage tank while still capable of providing substantially "instant" hot water when required.

[0014] The operating conditions, including maximum temperature and pressure, of the cogeneration system's first circuit are configured to be within the design range of the organic

working fluid. A controller can be incorporated to monitor and, if necessary, change operating parameters within the system. Switches, sensors and valves can be incorporated into the system to help the controller carry out its function. For example, to protect the expander from overspeeding during startup or shutdown transients, or low (or no) grid load, the controller can direct block and bypass valves to actuate, thereby forcing the superheated organic working fluid to bypass the expander. The controller may also integrate with user-determined conditions through the thermostat.

[0015] According to another aspect of the present invention, an indirectly-heated micro-CHP, including a heat source, first and second fluid circulating loops and an interloop heat exchanger, is disclosed. The indirectly-fired micro-CHP is advantageous in terms of system flexibility and maintainability. Multiple fluid-circulating loops are employed such that the heat source (for example, a burner) is provided to a second fluid circulating loop that is in thermal communication with, but fluidly isolated from, a first fluid circulating loop. The second fluid circulating loop includes piping used to convey a heat exchange fluid. This piping is preferably coiled and finned to maximize heat transfer between the heat source and the heat exchange fluid. At least one pump is used to circulate the heat exchange fluid. The second fluid circulating loop further contains a parallel set of sub-loops, one of which passes through a DHW heat exchanger to heat up municipal water, while the other passes through the interloop heat exchanger as an intermediary between the heat source and the organic working fluid flowing through the first fluid circulating loop. In addition to passing the organic working fluid through the interloop heat exchanger, the first fluid circulating loop includes a scroll expander connected to a generator, a SH heat exchanger, and a circulation pump. Upon the application of heat, the organic working fluid becomes superheated, then gets expanded in the scroll expander to turn the generator, thereby producing electrical power. The lower pressure, but still superheated organic working fluid leaving the scroll expander enters the SH heat exchanger, where another fluid, typically air or water, can be passed through and heated by the organic working fluid. This SH fluid is then circulated to radiators or similar space heating devices within a dwelling. The circulation pump

returns the condensed organic working fluid to the interloop heat exchanger, where it can repeat the process.

[0016] Optionally, a preheat device for the SH loop can be placed in heat exchange communication with the second fluid circulating loop such that additional SH may be effected. In addition, as with the previous aspect, the heat source may include a burner disposed within a combustion chamber-type container. The container may include an exhaust duct, an exhaust fan, and an exhaust gas heat exchanger disposed adjacent the exhaust duct. The exhaust gas heat exchanger can further include an exhaust gas recirculation device to further improve heat transfer from the exhaust gas. Residual heat that would otherwise be vented out the duct and to the atmosphere can be captured and rerouted to other parts within the system. For example, the exhaust gas heat exchanger may be integrated into the first sub-loop of the second fluid circulating loop in order to provide additional heating to the DHW heat exchanger.

[0017] According to yet another aspect of the present invention, a directly-fired cogeneration system configured to circulate an organic working fluid is disclosed. The directy-fired micro-CHP is advantageous in terms of system cost and simplicity. The system includes a piping loop that defines an organic working fluid flow path, an organic working fluid disposed in the piping loop, an evaporator disposed in the organic working fluid flow path, a burner in thermal communication with the evaporator such that heat transferred to the evaporator superheats the organic working fluid, a scroll expander disposed in the organic working fluid flow path such that the superheated organic working fluid passing through the scroll expander remains superheated upon discharge from the scroll expander, a generator operatively responsive to the scroll expander to generate electricity, a condenser, and a pump disposed in the organic working fluid flow path between the condenser and the evaporator. The condenser comprises a primary loop disposed in the organic working fluid flow path such that the primary loop is in fluid communication with the scroll expander, and a secondary loop in heat exchange relationship with the primary loop, where the secondary loop is configured to transfer at least a portion of the

heat contained in the organic working fluid passing through the primary loop to an external loop, such as a space heating device.

[0018] Optionally, the directly-fired micro-CHP system includes a controller, valves, combustion chamber and exhaust features similar to that of the previous aspects. Also, as with the previous aspects, the organic working fluid is preferably either a naturally occurring hydrocarbon (such as isopentane) or a halocarbon refrigerant, such as R-245fa. In addition, the heat source, which can be a burner, may be oversized to provide additional heat in variations of the system that do not employ a storage tank for DHW purposes. In this situation, the burner can be either larger, or a multi-staged device such that each stage is dedicated to a particular part of the external heating circuits, such as the SH or DHW circuits. Furthermore, the external heating circuits can be coupled to the cogeneration system from a single connection on the condenser such that bifurcated paths corresponding to the SH and DHW loops can both be accommodated.

[0019] According to still another aspect of the present invention, a micro combined heat and power system is disclosed. The micro combined heat and power system comprises an electricity generating loop and a connection to an external heating loop. The electricity generating loop includes a burner for raising the temperature of the organic working fluid such that the organic working fluid becomes superheated, a scroll expander to receive the superheated vapor such that the working fluid remains in a superheated state after passing therethrough, a generator operatively coupled to the scroll expander to produce electricity, a condenser disposed in fluid communication with the scroll expander and a pump to circulate the organic working fluid. The connection is disposed in the condenser, and is configured to place the external heating loop in thermal communication with the condenser. This external heating loop can be either a DHW loop, an SH loop, or both. As with the previous aspects of the invention, similar controller, combustion chamber and related features may be incorporated.

[0020] According to an additional aspect of the present invention, a system for the production of domestic hot water, space heat and electricity from a Rankine-based cycle with an organic

working fluid is disclosed. The system includes a substantially closed circuit fluid path configured to transport the organic working fluid therethrough, a burner configured to provide sufficient heat to superheat the organic working fluid, and a controller to regulate the operation of the system. The substantially closed circuit fluid path is at least partially defined by a coiled conduit configured to act as a heat transfer element for the organic working fluid, and includes as components a scroll expander, a generator, a condenser and a pump. The term "tube" can be used interchangeably with "conduit", as both describe a closed hollow vessel used for the transport of fluids. The burner is in thermal communication with the substantially closed circuit fluid path's coiled tube. The scroll expander is configured to accept the superheated organic working fluid. The condenser is configured to extract at least a portion of the heat remaining in the organic working fluid after the organic working fluid passes through the scroll expander. The pump pressurizes and circulates the organic working fluid.

[0021] According to yet an additional aspect of the present invention, an indirectly-fired cogeneration system comprising a heat source, a passive heat transfer element in thermal communication with the heat source, a first circuit, a generator and a second circuit is disclosed. The first circuit is configured to transport an organic working fluid, and is disposed adjacent an end of the passive heat transfer element such that heat transferred from the passive heat transfer element increases the energy content of the organic working fluid. The first circuit is made up of at least a scroll expander configured to receive the organic working fluid, a condenser in fluid communication with the scroll expander, and a pump configured to circulate the organic working fluid. The condenser is configured to transfer at least a portion of the excess heat contained in the organic working fluid to an external heating loop. As with the previous aspects, the generator is coupled to the scroll expander to produce electricity in response to motion imparted to it from the scroll. The second circuit is configured to transport a heat exchange fluid therethrough, and is disposed adjacent an end of the passive heat transfer element such that heat transferred therefrom increases the energy content of the heat exchange fluid. The second circuit is made up of at least a combustion chamber disposed adjacent the heat source such that exhaust gas can be removed. Details relating to the combustion chamber are similar to those discussed in

conjunction with the previous aspects, with the exception that one end of the passive heat transfer element (which is preferably a heat pipe) is disposed inside the combustion chamber so that such end absorbs heat from the heat source.

[0022] According to still another aspect of the present invention, a cogeneration system comprising a heat source, a passive heat transfer element in thermal communication with the heat source, and a first circuit is disclosed. The first circuit is configured to transport an organic working fluid, and is disposed adjacent an end of the passive heat transfer element such that heat transferred from the passive heat transfer element superheats the organic working fluid. The first circuit is made up of at least a scroll expander configured to receive the organic working fluid, a condenser in fluid communication with the scroll expander, and a pump configured to circulate the organic working fluid. A generator is coupled to the scroll expander to generate electricity in response to the expansion of the organic working fluid in the scroll. The condenser is configured to transfer at least a portion of the excess heat contained in the organic working fluid to an external heating loop. As with the previous aspect, the passive heat transfer element is preferably a heat pipe, and its integration into the combustion chamber is similar.

[0023] According to another aspect of the present invention, a method of producing heat and electrical power from a cogeneration device is disclosed. The method includes the steps of configuring a first circuit to transport an organic working fluid, superheating the organic working fluid with a heat source that is in thermal communication with the first circuit, expanding the superheated organic working fluid in a scroll expander, turning a generator that is coupled to the scroll expander to generate electricity, cooling the organic working fluid in a condenser such that at least a portion of the heat in the organic working fluid passing through the condenser is transferred to an external heating loop, using at least a portion of the heat that has been transferred to the external heating loop heat to provide space heat, and returning the organic working fluid exiting the condenser to a position in the first circuit such that it can receive additional heat input from the heat source.

[0024] Optionally, the method includes maintaining the organic working fluid in a superheated state through the expanding step. As an additional step, the method can selectively use at least a portion of the heat that has been transferred to the external heating loop to heat a domestic hot water loop. An alternative set of steps can be used to configure a second circuit to transport a heat exchange fluid to a DHW loop where the DHW loop is decoupled from the SH loop that is thermally coupled to the condenser. The second circuit is defined by a piping loop flow path that is in thermal communication with the heat source. The second circuit is in heat exchange communication with at least one domestic hot water loop, such as a heat exchanger or a water storage tank, for example. The second circuit is configured such that at least a portion of the heat that has been transferred to the heat exchange fluid will go to heat a fluid (such as water) in the domestic hot water loop. Preferably, the organic working fluid is superheated to about 10 to 30 degrees Fahrenheit above its boiling point in the superheating step, and is pressurized to a maximum pressure of about 200 to 450 pounds per square inch in the returning (pumping) step. In addition, the superheating step produces a maximum temperature of between about 250-350 degrees Fahrenheit in the organic working fluid. Moreover, the expanding step is conducted such that the electrical output of the generator is up to 10 kilowatts, while the cooling step is conducted such that the thermal output transferred to the external heating loop is up to 60 kilowatts. The heat source can either directly or indirectly fire the organic working fluid. An additional step may further include operating a set of valves configured to permit the organic working fluid to bypass the scroll expander upon a preset condition, which can be a grid outage, startup transient or shutdown transient.

[0025] According to another aspect of the present invention, a system for the production of electricity and space heat through the expansion of an organic working fluid in a superheated state is disclosed. The system comprises an organic working fluid, a flow path configured to transport the organic working fluid, a combustion chamber disposed in the flow path, a scroll expander disposed in the flow path to receive and discharge the organic working fluid in the superheated state, a generator operatively coupled to the scroll expander to produce electricity, a condenser in fluid communication with the scroll expander, and a pump to circulate the organic

working fluid through the flow path. The combustion chamber comprises a burner, a heat transfer element adapted to convey the organic working fluid adjacent the burner, and an exhaust duct to convey combustion products produced by the burner to the atmosphere. As with previous aspects, coupling between the condenser and an external heating loop can be used to effect heat exchange with an SH loop. In addition, system regulating devices, such as a controller, switches and valves may be employed, as can additional heat exchange devices that couple to the exhaust duct or the condenser, also discussed in conjunction with the previous aspects.

BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS

[0026] The following detailed description of the preferred embodiments of the present invention can be best understood when read in conjunction with the following drawings, where like structure is indicated with like reference numerals and in which:

~~FIG. 1 shows a schematic diagram of an integrated micro CHP system including a control subsystem according to the present invention;~~

~~FIG. 2 shows a schematic diagram of a PLC control subsystem according to one aspect of the present invention;~~

~~FIG. 3 is a start up/operating flow chart of the operating logic of one embodiment of the integrated micro CHP system according to the present invention;~~

~~FIG. 4 is a shutdown flow chart of the operating logic of the integrated micro CHP system according to one embodiment of the present invention;~~

~~FIG. 5 shows a schematic diagram of a sensor that may be used by a control subsystem of one embodiment of the present invention to sense the saturation pressure of an organic working fluid of an integrated micro CHP system;~~

~~FIG. 6 is a chart showing measured outside ambient temperature versus desired hydronic set point; and~~

~~FIG. 7 is a chart showing net power output at various hydronic supply temperatures, which varies approximately linearly with mass flow.~~

[0027] FIG. 1 shows a schematic diagram of an integrated micro-CHP system according to an embodiment of the present invention showing an indirectly-fired configuration with a storage tank and both SH and DHW capability;

[0028] FIG. 2 shows a schematic diagram of an integrated micro-CHP showing an indirectly-fired configuration with no storage tank and both SH and DHW capability;

[0029] FIG. 3 shows a schematic diagram of an integrated micro-CHP showing a directly-fired configuration with no storage tank and both SH and DHW capability;

[0030] FIG. 4 shows a schematic diagram of an integrated micro-CHP showing a directly-fired configuration with a storage tank and both SH and DHW capability;

[0031] FIG. 5 shows a schematic diagram of an integrated micro-CHP showing a directly-fired configuration with no storage tank and SH capability;

[0032] FIG. 6 shows the integration of a heat pipe into an indirectly-fired embodiment of the present invention, further highlighting a common heat exchanger for both SH and DHW;

[0033] FIG. 7 shows the integration of a heat pipe into a directly-fired embodiment of the present invention, further highlighting a common heat exchanger for both SH and DHW; and

[0034] FIG. 8 shows the details of an exhaust gas heat exchanger, including details of an exhaust gas recirculation device.

DESCRIPTION OF THE PREFERRED EMBODIMENT

~~With reference to the figures, an embodiment of the invention is illustrated incorporated into a number of the major components of an illustrated organic rankine cycle (ORC) micro-~~

~~cogeneration system used for the supply of domestic electrical power, domestic space heating (SH) water, and domestic hot water (DHW). Fluid supply lines connecting these components together are not drawn for ease of illustration. However, it is to be understood that the present control subsystems and control method of the present invention may advantageously be used with a number of micro cogeneration systems. Exemplary micro cogeneration systems are disclosed by co-owned U.S. Patent Application No. 09/998,705 filed November 30, 2001, the entire disclosure of which is herein incorporated fully by reference.~~

~~Referring initially to FIG. 1, a number of system components of an exemplary micro CHP system **100** is schematically shown, such as an expander **101,** a condenser (heat exchanger) **102**, feed pump **103,** heat source (boiler) **104**, and evaporator **105**. The expander **101** may be of any design, such as for example, a positive displacement expander and a scroll expander. In a direct fired system, the feed pump **103** circulates an organic working fluid (such as naturally occurring hydrocarbons or halocarbon refrigerants, not shown) through a loop at least defined by the fluidly connected expander **101**, condenser **102**, feed pump **103** and evaporator **105**. In an indirectly fired system, the feed pump **103** circulates the working fluid through a first or inner loop, and at least defined by fluidly connected expander **101**, condenser **102**, feed pump **103**, and a first loop portion of evaporator (heat exchanger) **105,** which is illustrated by dashed line**105a**. In such an arrangement, the first loop portion or interloop portion **105a** receives heat from a second loop portion of the evaporator or heat pipe which is directly heated by heat source **104**. More detailed information is provided in co-pending U.S. patent application No. 09/998,705 regarding directly and indirectly micro CHP system arrangements, to which reference is made.~~

~~Optionally, a desuperheater **106** may be included between the outlet of the expander **101** and the inlet of condenser/heat exchanger **102**. In this alternative embodiment, a regulated by pass valve **107** is used to circulate the working fluid exiting feed pump **103** through the coil loops (not shown) of the desuperheater **106** extracting heat energy from the vaporized working fluid exiting expander **101** before being circulating into evaporator **105** or interloop portion **105a**, if so configured. Since the size of the desuperheater is fixed, the amount of heat removed from the~~

~~working fluid vapor exiting expander **101** will vary with working fluid flow rate and temperature as the system modulates. However, it is to be appreciated that controllable by pass valve **107** permits the ability to control the flow of liquid working fluid through the desuperheater. By controlling the liquid working fluid flow, the amount of de superheating is controllable, and therefore the heat to power ratio of the micro CHP is controllable.~~

~~When maximum thermal output is required, by passing the desuperheater **106** will divert all the thermal energy to the hydronic fluid in condenser **102**. When maximum electrical power output is desired, directing all the liquid refrigerant coolant first to the desuperheater **106** will remove the maximum amount of energy from the working fluid vapor in the desuperheater prior to it entering the condenser **102**. By controlling the liquid working fluid flow rate between zero and 100 percent mass flow to the desuperheater **106**, the thermal and electrical outputs can be tailored to better match the space heating load and the electrical power load of the house or building.~~

~~A generator **108** (preferably induction type) is coupled to expander **101** such that motion imparted to it by expander **101** generates electricity. While the expander **101** can be any type, it is preferable that it be a scroll device. The scroll expander can be a conventional single scroll device, as is known in the art. The generator **108** is preferably an asynchronous device, thereby promoting simple, low cost operation of the system **100**, as complex generator speed controls and related grid interconnections are not required. An asynchronous generator always supplies maximum possible power without controls, as its torque requirement increases rapidly when generator **108** exceeds system frequency. The generator **108** can be designed to provide commercial frequency power, 50 or 60 Hz, while staying within close approximation (often 150 or fewer revolutions per minute (rpm)) of synchronous speed (3000 or 3600 rpm). The load on the expander **101** imposed by the grid ensures that mechanical speeds in the expander **101** are kept within its structural limits.~~

~~Inherent in a micro CHP (cogeneration) system is the ability to provide heat in addition to electricity. Excess heat, from both the heat source **104** and the expanded working fluid, can be transferred to external DHW and SH loops. The nature of the heat exchange process is preferably through either counterflow heat exchangers (e.g., for either or both the DHW and SH loops), or through a conventional hot water storage tank (e.g., for a DHW loop). In one embodiment, for example, a simple SH loop **109** may include fan **110** which provides for the drawing and blowing of air heated by heat source **104** to a space **111**. For DHW services, an external heating loop **112** (shown partially) may be coupled to condenser **102**. As an option, a preheat coil (not shown) can be inserted into the external heating loop **112** such that the hydronic fluid (typically water) flowing therethrough can receive an additional temperature increase by virtue of its heat exchange relationship with the heat exchange fluid flowing through a second circuit (not shown).~~

~~The hydronic fluid flowing through external heating loop or DHW system **112** is circulated with a conventional hot water pump **113**, and is supplied as space heat via a radiator or related device (not shown) to, for example, space **111**. As an example, hydronic fluid could exit the condenser **102** at about 50^{Θ} Celsius and return to it as low as 30^{Θ} Celsius. It will be appreciated by those of ordinary skill in the art that while the embodiments depicted in the figures show separate DHW and SH heat exchangers, it is within the spirit of the present disclosure that series, parallel, sequential and/or the same heat exchange configurations could be used. Additionally, although the capacity of the system **100** is up to 60kW, if desired, larger or smaller capacity units could be controlled by the herein disclosed control subsystem and method of the present invention.~~

~~Combustion chamber **114**, enclosing heat source **104**, includes at least a flumed exhaust duct **115** to the outside of the building, and an exhaust gas fan **116**. Heat at least to the evaporator **105** is provided by heat source or burner **104**, which is supplied with fuel by a gas train **117** having a shut off valve **118** and variable flow gas valve **119**.~~

~~Other devices may be used with the exhaust duct **115**, such as an exhaust gas recirculation device with an exhaust duct heat exchanger (not shown) which can be used to improve the thermal efficiency of the system **100** by lowering the temperature of the exhaust gas that is pulled away and vented to the atmosphere by fan **116**. The heat given up by the exhaust gas in an exhaust gas heat exchanger may be used to provide additional heat to other parts of the system **100**. More detailed information is provided by co-pending U.S. patent application No. 09/998,705 regarding exhaust gas recirculation devices with or without an exhaust duct heat exchangers, to which reference is made.~~

~~Block valve **120** and bypass valve **121** are situated in the organic working fluid flow path defined by piping **122** (of which bypass conduit **123** is part). In response to a no load condition (such as a system start up/shut down or grid outage) on the system, the superheated vapor in piping **122** is permitted to bypass around expander **101** through bypass conduit **123**, thereby avoiding overspeed of expander **101**. In this condition, the rerouted superheated vapor is fed into the inlet of condenser **102** or if so configured, desuperheater **106**. Under normal operating conditions, where there is a load on the system, the superheated vapor enters the expander **101**, causing the orbiting involute spiral wrap to move relative to the intermeshed fixed involute spiral wrap. As the superheated vapor expands through the increasing volume crescent shaped chambers of the expander **101**, the motion it induces in the orbiting wrap is transferred to the generator **108**, via a coupled shaft or an integral rotor/stator combination on the expander **101**.~~

~~Controller Sub-systems~~

~~As illustrated by FIG. 2, provided are controller sub-systems, generally indicated by **124**, to control the thermal and electrical output of integrated micro combined heat and power generation (M-CHP) systems, such as illustrated by FIG. 1. The controller sub-system **124** are microprocessor based, wherein the M-CHP system **100** uses a microprocessor controller **125**, which is preferably a programmable logic controller (PLC) or alternatively, any other conventional microcomputer, to monitor and control the operating conditions of system **100**. It is to be appreciated that a PLC based subsystem simplifies setup and troubleshooting and may accommodate programming modifications from future developments and refinements to such~~

~~micro CHP systems. In a preferred embodiment, the controller is an Allen Bradley MicroLogix 1500 PLC with a 1764 LRP Processor and includes other components, such as for example, shown below in Table 1 for it operation, wherein the controller 125 is programmed using conventional programming software and techniques. In a preferred embodiment, the controller **125** ladder logic may be programmed with Allen Bradley RS Logix 500 software, wherein a graphical maintenance program running Allen Bradley RS View on a laptop PC can be used for troubleshooting and setup in the field.~~

~~The controller subsystems **124** further include a plurality of process control sensors **126**, a plurality of control outputs **127**, a protection relay subsystem **128**, a system trip reset **129**, and the necessary user and service personnel interfaces, such a maintenance subsystem **130** and a data acquisition and communications subsystem **131**. Since the user and service personnel interfaces are conventional, for brevity no further discussion is provided.~~

~~For the purpose of controlling and monitoring the operation of cogeneration system **100**, the plurality of process control sensors **126** and control outputs (I/O devices) **127** are individually called out in Table 2 and shown in FIG. 1, by sensors S_0-S_{19}, control points A_1-A_{11}, and protection relays M_1-M_4, for the system at various points therewithin. As illustrated in FIG. 2, the sensors and I/O devices shown in Table 2 are interfaced with the controller **125** for control of the micro CHP system **100**. Additionally, the controlled and monitored parameter of sensors S_0-S_{19} and control points A_1-A_{12} are also listed in Table 2, as well as the preferred interface and type for each.~~

~~Table 1~~

~~Component~~	~~Mfg Part No~~	~~Description~~
~~Processor~~	~~1764LRP~~	~~MicroLogix 1500 Processor~~
~~Base~~	~~176424BWA~~	~~MicroLogix 1500 Base 24 I/O~~
~~Analog Voltage Input Module~~	~~1769IF4~~	~~Analog Voltage Input Module~~
~~Analog Voltage Output Module~~	~~1769OF2~~	~~Analog Voltage Output Module~~
~~Right end cap terminator~~	~~1769ECR~~	~~Right end cap terminator~~

~~Component~~	~~Mfg Part No~~	~~Description~~
~~Cable~~	~~1761CBLPM02~~	~~Cable~~
~~Thermocouple Input Module~~	~~1769 IT6~~	~~1769 IT6 Thermocouple Input Module~~
~~Real time clock~~	~~1764 RTC~~	~~Real time clock~~
~~Remote access modem~~	~~MICRORAD~~	~~Remote access modem~~

~~The information obtained from sensors **S_1-S_{19}** is used by the controller **125** to provide detailed system control through actuation and/or modulation of the various control points **A_1-A_{11}**. Such controlled and monitored parameters include pump speeds, gas flow rate, inlet and outlet temperatures and/or pressures to, from, and within the expander **101**, condenser **102**, evaporator **105**, combustion chamber **114**, desuperheater **106**, and throughout various points of the space heat, domestic hot water, and generation loops **109**, **112**, and **122**, respectively. Controlling and monitoring at least three of the above mentioned parameters (as will be discussed in later sections) enables setting the capacity of the system **100** at any instant in time, thereby permitting load following, using a variable capacity operation.~~

~~All of the pumps and fans **103**, **110**, **113**, and **116**, are responsive to input signals from controller **125** via the their associated control points, namely A_{10}, A_8, A_7, and A_{11}, respectively. The controller **125** uses appropriate program logic, as will be explained in a later section, to control the turning on/off the fans, running and varying the speeds of the pumps, and also opening and closing valves, such as for example, by pass valves **107**, **121**, shut off valves **118** and **120**, and variable flow gas valve **119**, in response to predetermined conditions. Such predetermined conditions include the demand for maximum heat or power output and an electric grid outage. Valves **107**, **118**, **119**, **120**, and **121** are responsive to input signals from controller **125** via the their associated control points, namely A_4, A_1, A_6, A_3, and A_2, respectively.~~

~~The controller **125** also monitors a number of additional safety controls and system protection devices, such as relays/contactor **132** (FIG. 1) of the alternator to grid having monitors **M1-M4** listed in Table 2. Additionally, the controller **125** monitors and reports the electrical trips to the feed pump, an oil pump, the hydronic pump, the blower, the gas valve, the expander bypass~~

~~valves, and any other electrically powered device in the system. Furthermore, the controller may monitor and activate a contactor of a crankcase heater for the power module via control point A_9 and switch the output of the generator 108 to the power grid via control point A_5.~~

~~In addition to controlling and monitoring system operations, the controller **125** possesses functions to optimize the performance, efficiency, and safety the system **100**. The functions of the controller **125** include automatic start-up and shut down, modulation control of hydronic supply temperature, monitoring and tripping on safety and abnormal operating parameters, and electrical grid connection interface. With reference also to FIGS. 3-4, these functions of the controller **125** are explained in further detail.~~

~~FIG. 3 is a flow chart of the operating logic of the system **100**. In operation, the controller **125** in one embodiment is programmed to operate the system in a quasi-steady state in response to a need for heat that is keyed to a specified hydronic supply temperature set point. This automatic operation is a heat load following mode. In other embodiments, such a call for heat, for example, may be from a thermostat, such as sensor S_0, which demands heat when the space temperature falls below a user set point, or according to an on-off timer, used for overnight shutdown. In the latter case, a timer would enable the thermostat to signal the controller **125** to initiate startup.~~

~~In the heat load following mode, the call for heat occurs whenever the controller **125** determines that the temperature differential between the actual hydronic supply temperature of the DHW system **112**, sensed via sensor S_2, and a desired hydronic supply temperature is greater the a predetermined value, such as, for example 0.5 to 5° Celsius.~~

~~Table 2~~

~~**Sensor I/O Device**~~	~~**Parameter**~~	~~**PLC Interface**~~	~~**Type**~~
~~S_0~~	~~Space Temperature~~	~~Digital Input~~	~~Thermostat Or Analog Signal Thermometer~~
~~S_1~~	~~Hydronic Condenser Inlet Temperature~~	~~Thermocouple Input~~	~~Thermocouple~~
~~S_2~~	~~Hydronic Supply Temperature~~	~~Thermocouple Input~~	~~Thermocouple~~
~~S_3~~	~~System Trip~~	~~Data Logger~~	~~Error Signal~~

~~S_4~~	~~Expander Inlet Temperature~~	~~Thermocouple Input~~	~~Thermocouple~~
~~S_5~~	~~Expander Inlet Pressure~~	~~Analog Input~~	~~Pressure Transducer (0-500 Psi)~~
~~S_6~~	~~Feed Pump Inlet Temperature~~	~~Thermocouple Input~~	~~Thermocouple~~
~~S_7~~	~~Feed Pump Inlet Pressure~~	~~Analog Input~~	~~Pressure Transducer (0-200 Psi)~~
~~S_8~~	~~Power Module Power Output~~	~~Digital Input~~	~~Amp/Watt Meter~~
~~S_9~~	~~Condenser Internal Pressure~~	~~Analog Input~~	~~Pressure Transducer (0-200 Psi)~~
~~S_{10-n10}~~	~~Evaporator Coil Temperature~~	~~Thermocouple Input~~	~~Thermocouple~~
~~S_{11}~~	~~Gas Flow~~	~~Digital Input~~	~~Flow Rate Meter~~
~~S_{12}~~	~~Expander Outlet Temperature~~	~~Thermocouple Input~~	~~Thermocouple~~
~~S_{13}~~	~~Feed Pump Set point~~	~~Digital Input~~	~~Flow Rate Meter~~
~~S_{14}~~	~~Heat Source High Limit~~	~~Thermocouple Input~~	~~Thermocouple~~
~~S_{15}~~	~~Flame Detection~~	~~Digital Input~~	~~Electro Optical~~
~~S_{16}~~	~~High Feed Level~~	~~Digital Input~~	~~Level Indicator~~
~~S_{17}~~	~~Low Feed Level~~	~~Digital Input~~	~~Level Indicator~~
~~S_{18}~~	~~Outside Temperature~~	~~Digital Input~~	~~Thermostat Or Analog Signal Thermometer~~
~~S_{19}~~	~~Hydronic Fluid Flow~~	~~Digital Input~~	~~Flow Rate Meter~~
~~A_1~~	~~Main Gas Valve~~	~~Relay Output~~	~~Solenoid Valve Actuator~~
~~A_2~~	~~Expander Bypass~~	~~Relay Output~~	~~Solenoid Valve Actuator~~
~~A_3~~	~~Expander Shutoff~~	~~Relay Output~~	~~Solenoid Valve Actuator~~
~~A_4~~	~~Desuperheater Bypass~~	~~Relay Output~~	~~Solenoid Valve Actuator~~
~~A_5~~	~~Power Module Run/Stop~~	~~Relay Output~~	~~Power Module Contactor~~
~~A_6~~	~~Evaporator Burner Firing Rate~~	~~Analog Output~~	~~Gas Proportioning Valve Actuator~~
~~A_7~~	~~Hydronic Pump Run And Speed~~	~~Relay Output Analog Output~~	~~Pump Variable Frequency Drive~~
~~A_8~~	~~Space Heat Fan Run/Stop~~	~~Relay Output~~	~~Motor Contactor~~
~~A_9~~	~~Crank Case Heater On/Off~~	~~Relay Output~~	~~Crank Case Heater Contactor~~
~~A_{10}~~	~~VFD - Feed Pump Run And Speed~~	~~Relay Output Analog Output~~	~~Pump Variable Frequency Drive~~
~~A_{11}~~	~~Forced Draft Fan Run/Stop~~	~~Relay Output~~	~~Motor Contactor~~
~~M_1~~	~~Net Failure~~	~~Digital Input~~	~~Protection Relay~~
~~M_2~~	~~Over/Under Voltage~~	~~Digital Input~~	~~Protection Relay~~
~~M_3~~	~~Over/Under Frequency~~	~~Digital Input~~	~~Protection Relay~~
~~M_4~~	~~Power Module Internal Temperature~~	~~Digital Input~~	~~Digital Thermometer~~

~~It is to be appreciated that the hydronic pump **113** operates continuously so there is always a flow through the DHW system **112** enabling the controller **125** to continuously monitor the actual supply temperature and return temperatures, via sensors S_2 and S_1, respectively. In other embodiments, the controller **125** may use the internal temperature of the condenser **102** and hydronic flow rate, via speed control of hydronic feed pump **113**, to determine a hydronic supply temperature using predetermined heat exchange values and/or algorithms.~~

~~The desired hydronic supply temperature is set by the controller **125** sensing the outdoor ambient temperature, via thermostat S_{18}, and correlating the sensed outdoor ambient temperature to a~~

~~desired hydronic supply temperature for the boiler. The correlation between outdoor ambient temperature and desired hydronic supply temperature may be, for example, according to the illustrated linear relationship shown by FIG. 6. In this illustrative embodiment, for example, on cold days, say -20°C ambient, the hydronic set point is 75°C, but on warm days, when only a little heat is needed, i.e., 20°C ambient, the set point is 25°C, and at a 0°C ambient the set point is 50°C. However, for other embodiments other scalar relationships between outdoor ambient temperature and the desired hydronic supply temperature may be used such as, for example, logarithmic, exponential, and other nonlinear functions. To avoid the influence of sunshine on cold days, a single measuring point on a north facing side of a building or home should be used for thermostat S_{18}.~~

~~Start-Up~~

~~In response to the call for heat in step **200**, the controller **125** is programmed to modulate the thermal output of the evaporator **105** in order to have the actual hydronic supply temperature match the desired hydronic set point for the given outdoor ambient temperature. Accordingly, the controller **125** in step **202** determines whether the evaporator **105** is already at its operating temperature via thermocouple sensor S_{10} or $S_{10'}$ (FIG. 1). Assuming the temperature of the evaporator **105** is below the minimum operating temperature (e.g., cold start up), the controller **125** verifies, and if necessary closes the expander shutoff valve **120** and opens the expander bypass valve **123** as shown in steps **204** and **206**, respectively. Once verified, the controller **125** checks to see if the burner is on via flame detector S_{15} and if not, purges the combustion chamber **114** with force draft blower **116** and activates the burner **104**, as shown at blocks **208** and **210**.~~

~~The burner **104** is activated by controller **125** opening the gas valve **118** and metering its flow with flow rate valve **119**, via actuator A_1, to nominally between 40 and 80% of full flow when ignition is expected. An igniter (not shown) is turned off by a timer and the gas valve is turned off when no flame is proven via flame detector S_{15}. The burner **104** will then typically be set to come on to about 50% of its capacity to warm up system **100**. If desired, an off the shelf combustion controller can initiate burner firing upon receiving an enabling signal from the~~

~~controller **125**. If so arranged, the combustion controller will provide flame and combustion air detection, light the burner, and provide high-temperature-limit protection for the evaporator **105**. The controller also drives a variable speed draft fan to provide the approximately correct air flow rate for the burner.~~

~~Next, the controller **125** checks to see if the evaporator working fluid level is in the desired operating range, via level sensors S_{16} and S_{17} in step **212** and if not, in step **214** will activate feed pump **103**. It is to be appreciated that both burner **104** firing and feed pump **103** flow may be controlled in part, and conventionally by room temperature and its user determined set point, as well as outdoor temperature, via sensor S_{18}. Additionally, feed pump **103** comes on to a speed predetermined by the controller **125** to coincide with the flow requirements established by the initial burner firing rate and the design response of the system **100**. In particular, the controller **125** runs feed pump **103** fast enough to keep the organic working fluid liquid level between level low sensor S_{17} and level high sensor S_{16}. Accordingly, the feed pump is turned on by the controller **125** at the appropriate time to keep the evaporator filled, but not over filled with working fluid.~~

~~When the system is operating, superheated working fluid is moving past sensor S_4, which is able to provide a valid signal to the controller **125** so the heat source or burner **104** firing rate and feed pump **103** flow can be adjusted for both the safe operation and needed output. However, when the system is just starting, the controller **125** must be given some initialized state which can be used as a safe operating condition until such time as working fluid is flowing past temperature sensor S_4.~~

~~It is desirable to have a minimum amount of working fluid flow during startup, so that the fluid heats up as rapidly as possible. However, some flow is needed to prevent local overheating of the fluid in evaporator **105**, and to provide the controller **125** with an indication that the burner **104** is indeed firing. Accordingly, the burner gas rate is set to provide the longest possible run time for the system, consistent with measured outdoor temperature and rate of change of indoor~~

~~temperature. Feed pump **103** operates to keep the evaporator **105** supplied with the working fluid at the factory preset value for temperature sensor S_4. When temperature sensor S_4 gets to about 50% of the thermostat set point, feed pump **103** speed is increased until the temperature reading in temperature sensor S_4 reaches its set point, at which time the burner **104** modulates for constant values of at least temperature sensor S_4, and the feed pump speed is modulated to maintain the desired hydronic supply temperature.~~

~~The controller will abort the start up sequence of steps **202-214**, if burner **104** fails to heat the evaporator **105** or if any of the parameters listed in Table 3 are exceeded. If the startup sequence fails, the controller **125** will attempt to re-start again for a maximum of three re-start attempts. If three re-start attempts are completed with no successful re-start, the controller **125** will be locked out from re-starting until manually reset, via trip reset **129** (FIG. 2). The re-start attempt counter will reset after a successful start or a manual reset.~~

~~After the startup sequence is complete the system **100** will go into run mode, wherein working fluid pressure is allowed to build and feed pump **103**, burner **104**, and evaporator **105** are controlled by controller **125** via two separate feed back loops.~~

~~At the appropriate time, the expander **101** is connected to receive the now superheated vapors of the working fluid, wherein the controller **125** in step **216** opens the cutoff or block valve **120** to expander **101** and after a short delay (i.e., 1 second), closes the bypass valve **121**. It is to be appreciated that liquid is prevented from entering into the expander by maintaining the evaporator's working fluid exit temperature at a fixed value for all operating conditions. As mentioned previously, in one embodiment, the evaporator exit temperature is set to operate at about 154°C (310°F), which has been found to give good overall system efficiency regardless of system load.~~

~~**Table 3**~~

~~**Parameter**~~	~~**Threshold**~~	~~**Fault Response**~~
~~Expander inlet temperature~~	~~340°F (171.1 °C)~~	~~Full shut down~~
~~Expander inlet pressure~~	~~420 psia (29.5 Kg/cm)~~	~~Full shut down~~
~~Feed pump inlet temperature~~	~~200°F (93.3°C)~~	~~Full shut down~~
~~Feed pump inlet pressure~~	~~200 psia (14 Kg/cm)~~	~~Full shut down~~
~~Protection relay trip~~	~~Suitable trip values~~	~~Power module shut down only~~
~~Power module temperature switch~~	~~300°F (148.9°C)~~	~~Power module shut down only~~
~~Startup sequence failure~~	~~3 attempts~~	~~Full shut down~~

~~Further protection to the scroll expander is provided during start up and while running by controlling the displacement of the feed pump, which in one embodiment is by feed pump speed, wherein the rate of change in feed pump speed is limited by an output ramp in the controller logic which provides time for the evaporator PID control to modulate the evaporator burner for steadier operation and to prevent over temperature of the expander inlet. In this control arrangement, a negative feed forward value will be added to the feed pump speed control variable when ramping down to better control temperature of the expander inlet.~~

~~The system **100** will warm up and quickly come to a near steady state operating point for the feed pump flow setting. After a short start up delay (i.e., 1 second) in step **218** wherein the controller verifies that the output of the generator is within set parameters via sensor S_8, the electrical output is then connected to the grid in step **220** via contactor **132** responding to control signal A_5 from the controller **125**. It is to be appreciated that there is no need to control the generator output as long as the generator output is less than the dwelling load. Preferably, connection of the generator **108** to the grid loads the alternator sufficiently to limit the speed to about 3150 RPM (50 hz application).~~

~~However, should the controller **125** sense a power module overspeed (e.g., loss of grid connection), it will quickly command the opening of a bypass loop **123** around the expander **101**.~~

~~Preferably, the sensor to detect expander overspeed is built into the power module. Furthermore, the controller **125** can be programmed with at least two options for dealing with the case when the generator output exceeds the dwelling load. The controller **125** can put the extra power on the grid or the extra power can be used for a resistance heater at condenser **102**.~~

~~Additionally, it is to be appreciated that the power module **108** will trip on over/under voltage, over/under frequency and loss of mains conditions detected by the protection relay **128** (FIG. 2). To prevent nuisance tripping of the heating system, only the power module **108** will trip on voltage and frequency grid disturbances. The controller **125** will log the fault and re-start the power module **108** after the over/under voltage or over/under frequency condition has dissipated. If a loss of mains occurs, the entire system **100** will shut down with all latches/relays reset to start ready condition. The controller **125** will then automatically re-start on the restoration of power if heat is called for by the thermostat/timer conditions.~~

~~In step **222**, the controller checks to see if the call heat has been satisfied via continuously reading sensor S_0 and/or S_{18}. If the hydronic supply temperature is not at the desired step point as checked in step **224**, the controller **125** in step **226** reads the outdoor ambient temperature via sensor S_{18} and uses a look up table or algorithm to establish the desired set point according to the linear relationship illustrated by FIG. 6. The controller **125** will then use the continuously updated desire set point to operate the evaporator **105** in a variable capacity mode by modulating the gas valve **119** on the burner **104** to maintain the actual hydronic supply temperature, sensed via sensor S_2, at the desired set point.~~

~~In step **228,** the controller **125** checks to see whether the expander inlet temperature via sensor S_4 is in a desired operating range. If not, then in step **230** the controller **125** modulates the burner fuel flow rate (heat input rate) to provide optimal thermodynamic performance and to prevent liquid from entering the expander. Additionally, in this quasi-steady state the controller **125** may control the hydronic pump **113** of the heating system at a speed that maintains the pressure~~

~~difference between the supply and return headers of the condenser at a preselected value for optimal thermodynamic performance of the hydronic heating system.~~

~~With the controller subsystems **124** operating the system **100** in the above described heat load following mode, it is to be appreciated that the system will operate for as many hours as possible during the heating season. The controller subsystems **124** will run the system **100** just hard enough to maintain the hydronic supply temperature at the correct value for the nominal heating load. When the system **100** operates at less than the maximum supply temperature, more power is generated than at the maximum temperature, the controller **125** can automatically and passively maximize the power which can be made and sold.~~

~~Further it is to be appreciated that in the above described heat load following mode, the greater the temperature differential error, the larger the thermal output response that will be initiated by controller **125** in order to achieve the desired hydronic supply temperature. As illustrated in FIG. 7, since the thermal output of the system varies approximately linearly with the mass flow of the system **100**, the controller **125** controls its thermal response by controlling both the displacement of the feed pump **103**, via a PID speed control, and the heat input to the evaporator **105**, via adjusting the gas flow rate to the burner **114** to maintain a 310°F (154.4°C) temperature into the expander.~~

~~In particular, the controller **125** is programmed to increase or decrease the mass flow rate in proportion to the system heating capacity required to match the actual and the desired set point for the hydronic supply temperature. For example, the desired set point of the hydronic supply temperature is 50°C for a given sensed outdoor ambient temperature of 0°C, if while operating in a quasi steady state, the actual hydronic supply temperature suddenly drops to 45°C (e.g., a door or window of the space or building is open to the outdoors), the controller **125** will increase the percentage of system mass flow to meet this heating demand. Should in this example after a predetermined time period the temperature error between the actual and desired hydronic supply temperatures decrease, the controller **125** will further increase the percentage of system mass~~

~~flow in order to meet this heating demand. Accordingly, it is to be appreciated that in normal operation the system **100** must operate at higher capacity to attain the higher hydronic supply temperatures required by colder weather and with the higher hydronic supply temperature the capacity of the indoor radiators/convectors increases to handle the added heat flow.~~

~~In run mode, the controller **125** monitors the operating parameters shown in Table 1 and performs a full or partial shut down if any of the parameters exceed preset thresholds. An individual trip event on the automatic trip settings is defined as a Level 1 trip. In the event of a Level 1 trip, the controller **125** will self reset and automatically attempt to re-start. If the re-start fails, the controller **125** will attempt to re-start again for a maximum of three re-start attempts. If the three re-start attempts are completed with no successful re-start, this case is defined as a Level 2 trip. In the event of a Level 2 trip, an alarm notification message will be sent by controller **125**, via the data acquisition and communications subsystem **131** and the controller **125** will be locked out from re-starting until manually reset, via trip reset **129** (FIG. 2). The re-start attempt counter will reset after a successful start or a manual reset.~~

~~Shutdown~~

~~Normal shutdown of the micro-CHP will occur if the outdoor thermostat signal indicates that the outside temperature has gone above the temperature that corresponds to the lowest hydronic supply temperature set point and remains for 30 minutes. The system will re-start when the thermostat signal indicates that the outside temperature is below the normal startup temperature. The controller **125** will also perform the normal shutdown sequence when switched off manually by the user or service personnel.~~

~~In the former case, when the system load falls below about 30 to about 40% of a full load and 30 minutes has expired, the controller **125** is programmed to shutdown the system **100** and cease making both heat and power to ensure economical use of the m-CHP system. Since the hydronic pump is kept running at all times, even at a low flow rate, the controller **125** continuously monitors the error signal between the hydronic actual and set point values. When this error is~~

~~large enough, (i.e. the actual temperature is below the set point by a preselected value) the controller starts the system for another on cycle. Should the controller also sense that during operation of the system at the minimum system mass flow, the actual supply temperature begins to exceed the set point, it is programmed also to shutdown the system when this error exceeds a predetermined value. In either of these conditions of heat load following mode, or upon receiving a heat satisfied message from a space thermostat, if not configured in the heat load following mode, the controller **125** will follow a normal shutdown procedure according to the program control logic illustrated by FIG. 4.~~

~~In step **300**, normal shutdown begins with the controller **125** turning off burner **103**. After a time delay in step **303** (i.e., 15 seconds), the controller **125** will shutdown (disconnect) the power module **108**, open the bypass valve **121** and close the shutoff valve **120** to the expander **101** in step **304**. After another time delay (i.e., 5 seconds) in step **306**, the controller **125** stops the feed pump **103** in step **308**. After another time delay (i.e., 60 seconds) in step **310**, the controller **125** may slow the hydronic pump **113** to its minimum speed, completing a normal shutdown of the system **100**.~~

~~However, in step **300** the time delay may be conditioned on available reservoir heat energy in heating system **112**, wherein if desired, the hydronic pump **113** may run on to deliver residual heat in the hydronic system to the radiators until below a certain set point. In such a case, the controller **125** will initiate a partial shutdown if the hydronic supply temperature exceeded the set point by 5°C for a period of 30 minutes when the hydronic set point is at the minimum setting. The evaporator **105**, feed pump **103**, expander **101**, and power module **108** will shutdown while the hydronic pump will continues to run. When the hydronic supply temperature error signal reaches – 0.5 to 5°C, the system will re-start.~~

~~Data will be logged using the logging capability of the controller **125**. Preferably, a remote access modem will interface with the controller **125** to download any system data. The modem may be self dialing and use either land line or cellular service. In the event of a Level 2 trip, the~~

~~controller may use the modem to send an alarm notification message to service personnel. Data points logged by controller **125** are shown in Table 4.~~

~~Evaporator Heat Control~~

~~In normal operation, no control of the evaporator heat rate is required as the ideal heat input rate is set by a given mass flow rate out of the feed pump. If the heat input rate is greater than the ideal rate, then the evaporator outlet fluid will be more superheated than desired. This will lead to increasing evaporator pressure until the density at the expander inlet is sufficient to provide a match between expander and feed pump mass flow rate. Thus, the results of excessive heat input rate will be excessive evaporator pressure and evaporator outlet superheat.~~

~~If there is too little evaporator heat input for a given feed pump flow rate, evaporator pressure will be reduced, and some liquid may be admitted to the expander **101**. Liquid admission to the expander **101** is not expected to result in damage to the expander because of the increasing chamber volume at all times. However, partial liquid admission is likely to result in reduced pressure ratio across the expander **101** and, therefore, less thermodynamic work by the expander. Accordingly, to obtain a better match between the evaporator heat input rate and the feed pump flow rate over all operating conditions, better control over the evaporator heat output is desired.~~

~~The control concept involves sensing the level of superheat at the evaporator outlet and maintaining this level in the range of 20°F to 30°F (-6°C to -1°C). This will tend to minimize the evaporator pressure while maintaining some margin from liquid entering the expander **101**. The evaporator pressure will then float until the density at the expander inlet is such that the expander and feed pump flow rates match.~~

~~Table 4~~

~~**Channel**~~	~~**Parameter**~~	~~**Signal Type**~~
~~1~~	~~Hydronic condenser inlet temperature~~	~~Thermocouple~~
~~2~~	~~Hydronic condenser outlet temperature~~	~~Thermocouple~~

~~Channel~~	~~Parameter~~	~~Signal Type~~
~~3~~	~~Hydronic fluid flow~~	~~Analog~~
~~4~~	~~Expander inlet temperature~~	~~Thermocouple~~
~~5~~	~~Expander inlet pressure~~	~~Analog~~
~~6~~	~~Feed pump inlet temperature~~	~~Thermocouple~~
~~7~~	~~Feed pump inlet pressure~~	~~Analog~~
~~8~~	~~Power module power output~~	~~Analog~~
~~9~~	~~Evaporator inlet temperature~~	~~Thermocouple~~
~~10~~	~~Gas flow~~	~~Analog~~
~~11~~	~~Expander outlet temperature~~	~~Thermocouple~~
~~12~~	~~Feed pump speed (drive frequency)~~	~~Analog~~
~~13~~	~~Protection relay trip~~	~~Digital~~
~~14~~	~~Level 1 trip failure~~	~~Digital~~
~~15~~	~~Level 2 trip failure (failed to re-start)~~	~~Digital~~

~~The superheat exiting the evaporator **105** may be sensed by either measuring the evaporator outlet temperature and saturation temperature, measuring the evaporator outlet temperature and outlet pressure, or sensing the saturation pressure. The first approach requires measuring the temperatures of the working fluid in its saturated state at the evaporator inlet with inlet temperature sensor S_{10} (indirectly fired) or $S_{10'}$ (directly fired) and also in its vapor state such as with expander inlet temperature sensor S_4. The difference between the temperatures is then the superheat. For an improved superheat sensing, in addition to the temperature, expander inlet pressure sensor S_5 may be used to sense the evaporator outlet pressure. The controller **101** may then use a look up table and/or calculation to determine the saturation temperature from the pressure and temperature readings, wherein the superheat is the difference between the measured temperature and the computed saturation temperature. Furthermore, pressure sensor S_5 can also provide a safety feature to protect both the evaporator **105** and the expander **101** from potentially harmful overpressure.~~

~~The third approach is illustrated by FIG. 5, wherein the evaporator outlet pressure is applied to one side of a diaphragm **500** of a superheat sensing device **501**. The other side of the diaphragm sees the same working fluid, but from a sensing bulb **502** at the same temperature as the~~

~~evaporator outlet gas. The pressure of this latter side is the saturation pressure corresponding to the evaporator outlet temperature. If there is positive superheat, the pressure from the sensing bulb **502** will exceed the evaporator outlet pressure and the diaphragm **500** will compress a spring **504** by an amount proportional to the degree of superheat. A potentiometer (or other position sensor) **506** attached to the diaphragm **500** then provides an electrical output to the controller **125** proportional to the degree of superheat. The controller **125** can then use the output sign from the superheat sensing device **501** to minimize the evaporator pressure while maintaining some margin from liquid entering the expander **101.**~~

~~Although some aspects of the present invention are identified herein as preferred or particularly advantageous, it is contemplated that the present invention is not necessarily limited to these preferred aspects of the invention. For example, an alternative mode of providing heat following control may be as follows. As before, this alternative heat following mode uses an outdoor temperature sensor (i.e., sensor S_{18}) to determine the set point of the hydronic supply temperature. Controller **125** after sensing the actual hydronic supply temperature (i.e., via sensor S_2), adjusts the burner firing rate, via the gas valve **118** and/or **119**, via actuator A_1, to bring the actual hydronic supply temperature into line with the desired set point. If the actual hydronic supply temperature is too low, the controller **125** will increase the firing rate, and if the temperature is too low , the controller **125** will decrease the firing rate.~~

~~Further, in this alternative heat following mode, to maintain the evaporator outlet temperature of the superheated working fluid vapors within a desired operating parameter, such as, for example 310ºF (154.4ºC), the controller **125** will adjust the feed pump flow rate and/or speed of the organic working fluid past the burner in order to control the evaporator exit temperature of the superheated vapor. If the exit temperature of the superheated working fluid vapors from the evaporator is above its desire operating parameter, the controller **125** will increase the working fluid's flow rate. Should the exit temperature of the superheated working fluid vapors from the evaporator be less than its desired operating parameter, the controller **125** will decrease the working fluid's flow rate.~~

[0035] Referring initially to FIG. 1, one embodiment of the micro-CHP system **1** is an indirectly-heated, dual-loop system that includes a first (or primary) circuit **100** and a second circuit **150**. An advantage of the indirectly fired system is that first circuit boiler (or evaporator) conduit overheating and subsequent burn-out is avoided. First circuit **100** includes a expander **101**, a condenser **102**, a pump **103** and one portion of interloop heat exchanger **104**. An organic working fluid (such as naturally-occurring hydrocarbons or halocarbon refrigerants, not shown) circulates through the loop defined by the fluidly-connected expander **101**, condenser **102**, pump **103** and interloop heat exchanger **104**. Piping **110** is used to connect the various components of first circuit **100**, whereas the pump **103** provides the pressure to supply the organic working fluid to the interloop heat exchanger **104**, thereby completing the first circuit **100**. A generator **105** (preferably induction type) is coupled to expander **101** such that motion imparted to it by expander **101** generates electricity. While the expander **101** can be any type, it is preferable that it be a scroll device. The scroll expander can be a conventional single scroll device, as is known in the art. An oil pump **108** is used to provide lubricant to the scroll. The presence of oil helps to establish a seal between the intermeshed stationary and orbiting wraps that make up the scroll's crescent-shaped chambers (not shown). A level indicator switch **120** with level high **120A** and level low **120B** indicators is placed at the discharge of condenser **103**. Controller **130** is used to regulate system operation. It senses parameters, such as organic working fluid temperatures, at various points within the first circuit and level information taken from the level indicator switch **120**. Through appropriate program logic, it can be used to open and close valves (not presently shown) in response to predetermined conditions, such as an electric grid outage. The generator **105** is preferably an asynchronous device, thereby promoting simple, low-cost operation of the system **1**, as complex generator speed controls and related grid interconnections are not required. An asynchronous generator always supplies maximum possible power without controls, as its torque requirement increases rapidly when generator **105** exceeds system frequency. The generator **105** can be designed to provide commercial frequency power, 50 or 60 Hz, while staying within close approximation (often 150 or fewer revolutions per minute (rpm)) of synchronous speed (3000 or 3600 rpm).

[0036] An external heating loop **140** (shown preferably as an SH loop) can be coupled to first circuit **100** via connectors (not shown) on condenser **102**. As an option, a preheat coil **145** can be inserted into the external heating loop **140** such that the hydronic fluid (typically water) flowing therethrough can receive an additional temperature increase by virtue of its heat exchange relationship with heat exchange fluid flowing through second circuit **150** (discussed in more detail below). The hydronic fluid flowing through external heating loop **140**, is circulated with a conventional pump **141**, and is supplied as space heat via radiator **148** or related device. As an example, hydronic fluid could exit the condenser **102** at about 50^{O} Celsius and return to it as low as 30^{O} Celsius. The capacity of the system **1** is up to $60kW_t$; however, it is within the scope of the present invention that larger or smaller capacity units could be utilized as needed. Inherent in a micro-CHP (cogeneration) system is the ability to provide heat in addition to electricity. Excess heat, from both the heat source and the expanded working fluid, can be transferred to external DHW and SH loops. The nature of the heat exchange process is preferably through either counterflow heat exchangers (for either or both the DHW and SH loops), or through a conventional hot water storage tank (for a DHW loop). It will be appreciated by those of ordinary skill in the art that while the embodiments depicted in the figures show DHW and SH heat exchangers in parallel (and in some circumstances being supplied from the same heat exchange device, shown later), it is within the spirit of the present disclosure that series or sequential heat exchange configurations could be used.

[0037] Second circuit **150** includes two parallel sub-loops **150A, 150B**. Heat to the two parallel sub-loops **150A, 150B** is provided by a burner **151**, which is supplied with fuel by a gas train **152** and variable flow gas valve **153**. Piping **160** (which makes up the parallel sub-loops) passes through a combustion chamber **154**, which is where the heat from the combustion of fuel at burner **151** is given up to the heat exchange fluid (not shown) that flows through piping **160**. Piping **160**, which includes a finned tube portion **161** disposed inside the combustion chamber **154**, branches out into the first parallel sub-loop **150A**, which transports the heat exchange fluid that has been heated in combustion chamber **154** to interloop heat exchanger **104** in order to give up the heat to organic working fluid flowing through first circuit **100**. Block valves (not shown)

could be used to regulate flow between the sub-loops; however, by idling the pump of the inactive sub-loop, significant flow in that sub-loop is prevented without the need for additional valving. The second parallel sub-loop **150B** transports the heat exchange fluid to DHW heat exchanger **180** in order to heat up domestic hot water. One side of domestic hot water heat exchanger **180** (which can be a water storage tank) includes coil **180A** configured to transport the heat exchange fluid, and another side, the shell **180B**, to transport domestic hot water (not shown) from a cold water inlet **191A**, past coil **180A** and to DHW outlet **191B**. Typically, the cold water comes from either a well or a city/municipal water supply. Similarly, temperature sensor **171B** can detect the temperature of the DHW coming out of the DHW heat exchanger **180**. This sensor can also be linked to a controller **130** (discussed in more detail below). Combustion chamber **154** includes an exhaust duct **155**, an exhaust gas recirculation device **156** with exhaust duct heat exchanger **157**, and fan **158**. It will be appreciated by those skilled in the art that although the fan **158** is preferably shown as an induced-draft fan, it could also be a forced-draft fan, if properly located relative to the combustion chamber **154**. Temperature sensor **171A** is placed at the combustion chamber **154** outlet for the second circuit **150** to measure the temperature conditions of the heat exchange fluid, in a manner similar to that of temperature sensor **171B**. Second circuit pumps **185A**, **185B** are used to circulate heat exchange fluid through the second circuit **150**, with pump **185B** circulating heat exchange fluid through DHW heater **180** and pump **185A** circulating heat exchange fluid through interloop heat exchanger **104**. The exhaust duct heat exchanger **157** and an exhaust gas recirculation (EGR) device **156** accept hot exhaust gas from the burner **151** and recirculate it in an internal heat exchange process, thereby lowering the temperature of the exhaust gas that is pulled away and vented to the atmosphere by fan **158**. The heat given up by the exhaust gas in the exhaust gas heat exchanger **157** is used to provide additional heat to other parts of the system **1**. In the present figure, this additional heat is used to increase the temperature of the heat exchange fluid flowing in second circuit **150**.

[0038] A controller **130**, which could be a programmable logic controller (PLC) or conventional microcomputer (not shown), can be used to provide detailed system control. All of the pumps

can be configured to be variable-speed, and are responsive to input signals from controller **130**. Upon receipt of a signal for heat, the burner **151** ignites the fuel, while the proper circulating pump **185B** or **185A** is energized. For DHW, flow switch **190**, in conjunction with temperature sensor **171B**, provide inputs to controller **130**. Flow switch **190** selects DHW mode, where the DHW set point is coupled to temperature sensor **171A**. The burner gas flow and DHW pump **185B** flow are regulated to provide the desired temperature at **171B** according to the temperature preset by the user on the DHW thermostat (not shown).

[0039] When the system is operating, heated heat exchange fluid is moving past sensor **171A**, which is able to provide a valid signal to the controller **130** so the burner **151** firing rate and pump **185B** flow can be adjusted for both safe operation and the needed output. However, when the system is just starting, the controller **130** must be given some initialized state which can be used as a safe operating condition until such time as heat exchange fluid is flowing past temperature sensor **171A**. It is desirable to have a minimum amount of heat exchange fluid flow during startup, so that the fluid heats up as rapidly as possible. However, some flow is needed to prevent local overheating of the fluid in the combustion chamber **154**, and to provide the controller **130** with an indication that the burner **151** is indeed firing. The gas rate is set to provide the longest possible run time for the system, consistent with measured outdoor temperature and rate of change of indoor temperature. Pump **185B** operates to keep the combustion chamber **154** supplied with the heat exchange fluid at the factory-preset value for temperature sensor **171A**. When temperature sensor **171A** gets to about 50% of the thermostat set point, the pump **185B** speed is increased until the temperature reading in temperature sensor **171A** reaches its set point, at which time the burner **151** and pump **185B** modulate for constant values of temperature sensors **171A** and **171B**. When the flow switch **190** indicates zero flow, the burner **151** and pump **185B** cease operation. A small expansion tank (not shown) can be placed in the second circuit **150** to allow for differential thermal expansion at moderately high pressures of the heat exchange fluid.

[0040] When the user desires heat, as indicated by the room thermostat (not shown) the burner **151** comes on to about 50% of its capacity to warm up system **1**. Pump **185A** comes on to a speed predetermined to coincide with the flow requirements established by the initial burner firing rate and the design response of the system. The controller **130** responds to the user demand for heat, and the owner selected set point for room temperature. Burner **151** firing and pump **185A** flow are controlled in part, and conventionally by room temperature and its set point, as well as outdoor temperature (sensor not shown). The first circuit pump **103** runs fast enough to keep the organic working fluid liquid level between level low **120B** and level high **120A** switch settings. The controller **130** instructs the pump **103** to start or speed up when the organic working fluid liquid level rises above the level **120A**, and stopping when the level goes below level **120B**, for example.

[0041] The length of finned tube portion **161** of piping **160** that is inside the combustor **154** can be minimized by carefully selecting pumps, control points, and conduit size. Referring now to FIG. 8 in conjunction with FIG. 1, details of the EGR device **156** for micro-CHP system **1** is shown. In essence, the EGR device **156** functions in conjunction with the exhaust duct **155** and is an integral part of exhaust gas heat exchanger **157**. The hot exhaust gas stream is directed axially through EGR device **156**, which is preferably placed between burner **151** and exhaust duct **155**. An annular recirculation duct **156B**, passes some of the exhaust gas in a counterflow fashion until it is reinjected at inlet **156A**. The walls of the EGR device **156** are cooled by the heat exchange fluid that passes through the duct heat exchanger **157**, and as a result, the recirculation gas entering at inlet plane **156A** is partially cooled. This tempered gas stream leaving at plane **156B** enters the second heat transfer section defined by finned tube portion **161** of second circuit piping (not presently shown), in which additional cooling of the gas occurs. In a more compact arrangement, the inner annular duct of the EGR device **156** would be replaced by an array of fine tubes (not shown), each having a flow inducer for hot gas at the inlet end. While such an approach would involve the use of a larger amount of fluid, which would increase the response time of the system, significant benefits could be realized, including the application of the EGR device **156** to an evaporator where an organic working fluid is used such that the

fluid is never exposed to the full temperature of the exhaust gas, and the final heat recovery is not reduced by any form of added flue gas dilution, especially cool air. The primary benefit of the EGR device **156** is that levels of harmful gaseous by-products (such as NO_x) are reduced. An additional benefit of the EGR device is that by reducing the highest temperature that the finned tube portion **161** is exposed to, simpler components that will have lower cost yet which can attain the same long life of more costly materials can be used.

[0042] Referring next to FIG. 2, an alternate embodiment of the indirectly-fired micro-CHP system **2** is shown. Here, the second circuit **250** does not encompass parallel sub-loops. Instead, a single loop is routed directly from combustion chamber **254** to interloop heat exchanger **204**. DHW capability, which was provided by second sub-loop **150B** in the embodiment shown in FIG. 1, is now integrated into the external heating loop **240**. This external loop, that services both DHW and SH, can be bifurcated after coupling to the condenser **202**, with valves **247A, 247 B** operating to supply SH radiators **248** or DHW heat exchanger **280** as needed. DHW heat exchanger **280** can be either a water tank to store hot water (as discussed in conjunction with the previous aspect), or a dual-pass counterflow heat exchange device. After the fluid (typically water) passes through either or both SH and DHW heat exchangers, it is circulated through heating loop **240** back to the condenser **202** to start its cycle again. Prior to entry into the condenser **202**, the fluid can be preheated by passing it thermally adjacent second circuit **250** in a preheat device **245**.

[0043] Referring now to FIGS. 3 and 4, a directly-fired micro-CHP system is shown. This system has the advantage of being simpler in construction, with attendant lower cost. In the present embodiment, the system **3** does not include a second circuit. The interloop heat exchanger of the previous embodiments, which acted as the heat source for the previous embodiment first circuits, is replaced by a combustion chamber **304**, where both the burning of fuel, through gas train **352**, valve **353** and burner **351**, and the evaporation of the organic working fluid takes place. As with the previous embodiments, the organic working fluid is superheated. Generator **305**, as with the previous embodiments, is asynchronously tied to a load,

preferably on the customer/user side of the electric meter, which is typically the power grid. The load on the scroll expander **301** imposed by the grid ensures that mechanical speeds in the scroll **301** are kept within its structural limits. Block valve **307A** and bypass valve **307B** are situated in the organic working fluid flow path defined by piping **310** (of which conduit **361** is part). These valves respond to a signal in controller **330** that would indicate if no load (such as a grid outage) were on the system, allowing the superheated vapor to bypass around the expander, thereby avoiding overspeed of scroll **301**. In this condition, the rerouted superheated vapor is fed into the inlet of condenser **302**. Under normal operating conditions, where there is a load on the system, the superheated vapor enters the scroll expander **301**, causing the orbiting involute spiral wrap to move relative to the intermeshed fixed involute spiral wrap. As the superheated vapor expands through the increasing volume crescent-shaped chambers, the motion it induces in the orbiting wrap is transferred to the generator **305**, via a coupled shaft or an integral rotor/stator combination on the scroll **301**. Depending on the type of oil used in the system (such as whether the oil is miscible or immiscible with regard to the organic working fluid), scroll **301** may preferably include an oil pump **308** to circulate oil present in the scroll from the superheated vapor. The workings of the exhaust duct **355** and fan **358** are similar to that of the previous aspect; however, the present EGR device **356** and exhaust duct heat exchanger **357**, rather than providing additional heat to a heat exchange fluid flowing through the second circuit **150, 250** of the previous embodiments, can be used to provide supplemental heat to various locations within the system **3**. For example, additional heat can be added to the organic working fluid coming out of pump **385**, shown at point **A**. Similarly, it can be used to add heat to the external heating loop **340** at points **B** or **C**. Precise location of the heat exchange points **A, B** or **C** would be determined by the nature of the organic working fluid and its properties. Note that DHW heat exchanger **380** can be configured as a conventional dual-pass counterflow heat exchanger, or as a water storage tank, as discussed in the previous aspects. In situations where no (or a small) storage tank is being used (such as, for example, when space is at a premium), then in order to provide fast-responding DHW, additional heat generation may be required. One approach is to use a larger or multiple-stage burner (not shown). This could provide rapid response times to the instant or near-instant demands associated with DHW uses (such as showers, baths and hot tap

water). Referring with particularity to FIG. 4, a variation on the directly-fired micro-CHP of FIG. 3 is shown. In this case, the system **4** specifically includes a storage tank **480**. This approach allows the inclusion of DHW capability without having to resort to increased burner capacity. In addition, power to a storage tank heating element **480C** can be provided directly off generator **405**. In addition, trade-offs between the size of the storage tank **480** and the size or number of burner **451** can be made to best suit the functionality and packaging/volume requirements of the system.

[0044] Referring now to FIG. 5, a directly-fired micro-CHP system **5** is shown. This represents the most simplistic system, in that it is geared toward the exclusive generation of electricity and SH. By not including DHW capability, a storage tank can be avoided without sacrificing system functionality or requiring augmented burner capacity. In other respects, this system is similar to that of the previous directly-fired embodiments, including operation of the heat source componentry **551, 552** and **553**, exhaust componentry **555, 556, 557** and **558**, organic working fluid flow path componentry **501, 502, 503, 504, 507A,B** and **508**, generator **505**, and sensing a controlling apparatus **520, 530**.

[0045] Referring now to FIGS. 6 and 7, a variation on the indirectly-fired and directly-fired cogeneration systems of the previous aspects is shown. Referring with particularity to FIG. 6, a passive heat transfer element, preferably in the form of a heat pipe **675**, can be disposed between the first circuit **600** and the second circuit **650** to effect heat exchange between those circuits and the heat source. Referring with particularity to FIG. 7, heat pipe **775** is disposed within the flow path of the first circuit, which also includes scroll expander **701**, condenser **702** and pump **703**. In either configuration, the heat pipe is an evacuated and sealed container that contains a small quantity of working fluid, such as water or methanol. When one end of the pipe (typically referred to as the evaporator end) is heated, the working fluid rapidly vaporizes, due in part to the low internal pressure of the fluid. The vapor travels to the lower-pressure opposite end (typically referred to as the condenser end), giving up its latent heat. Preferably, gravity or capillary action allows the condensed fluid to move back to the evaporator end, where the cycle can be repeated.

When the fluid has a large heat of vaporization, a significant amount of heat can be transferred, even when the temperature differences between the opposing ends is not great. In other regards, the operation of the systems is similar to that of the previous aspects.

[0046] Referring now to FIG. 8, details of the exhaust duct heat exchanger **157** and the exhaust gas recirculation device **156** are shown. The combustion chamber **154** (not presently drawn to scale) encases enough of the heat source apparatus, including burner **151**) to ensure that the exhaust gas and related combustion products are entrained into the exhaust duct **155** such that they can be vented to the atmosphere. An induced draft fan (shown elsewhere) can be used to ensure thorough venting of the combustion products. The exhaust gas recirculation device **156** is a co-annular duct that takes the exhaust gas leaving the region around burner **151** through the inner annulus **156A**, and doubles back a portion of the gas to flow in the outer annulus **156B**. During the time that the portion of the gas that is recirculating through the outer annulus **156B**, it is giving up some of its heat to the exhaust duct heat exchanger **157**, which is shown as a coiled conduit. From here, the coiled conduit of the heat exchanger **157** can be routed to other locations (shown elsewhere) in the system, where it can then be used to provide supplemental heat.

[0047] Having described the invention in detail and by reference to preferred embodiments thereof, it will be apparent that modifications and variations are possible without departing from the scope of the invention defined in the appended claims.

[0048] We claim:

~~1. A control subsystem for governing the operation of a cogeneration system configured to operate with an organic working fluid, said system having a plurality of functional devices including at least a heat source, an expander having operatively coupled to a generator to produce electricity, a condenser in fluid communication with said expander and adapted to heat a hydronic heating system, and a pump configured to circulate said organic working fluid from said condenser through piping in thermal communication with said heat source such that heat transferred therefrom superheats said organic working fluid to provide superheated organic working fluid vapors to said expander, said control subsystem comprising a programmed processor operatively coupled at least to said pump and said heat source, and adapted to operate said pump and said heat source in response to a call for heat, causing said organic working fluid in said piping in thermal communication with said heat source to be superheated and provided to said expander.~~

~~2. The control sub-system according to claim 1, wherein said functional devices of the cogeneration system further include a power grid contactor adapted to connect the electrical power output of said generator with a power grid and an electrical load, wherein said controller is further operatively coupled to said generator and said contactor for receiving and monitoring electrical power output from said generator, and connecting/disconnecting said electrical power output with said electrical load.~~

~~3. The control subsystem according to claim 1, further comprising a plurality of sensors each providing a sensor signal indicative of a parameter of the cogeneration system, said controller coupled to said plurality of sensors and adapted to at least monitor the operation of said functional devices of the cogeneration system.~~

~~4. The control subsystem according to claim 3, further comprising an operator interface coupled to said controller enabling modification of the operation of said controller in monitoring said functional devices and operating characteristics of the cogeneration system through entry of information via said operator interface.~~

~~5. The control subsystem according to claim 4, wherein said operator interface includes a data acquisition and communications subsystem enabling data logging and reporting of said operating characteristics of said functional devices of the cogeneration system.~~

~~6. The control subsystem according to claim 5, wherein said a data acquisition and communications subsystem includes a modem for reporting at least alarm conditions of said cogeneration system.~~

~~7. The control subsystem according to claim 3, wherein said parameters include at least three of condenser inlet temperature, condenser outlet temperature, hydronic fluid flow, hydronic supply temperature, expander inlet temperature, expander inlet pressure, feed pump inlet temperature, feed pump inlet pressure, power module power output, evaporator inlet temperature, gas flow, expander outlet temperature, outdoor ambient temperature, feed pump speed (drive frequency), protection relay trip, level 1 trip failure, and level 2 trip failure (failed to re-start).~~

~~8. The control sub-system according to claim 3, further comprising a plurality of control points, said controller being operatively coupled to said control points of said cogeneration system associated with functional devices thereof, said controller responding to said sensor signals to control said functional devices to thereby vary operating conditions of the cogeneration system.~~

~~9. The control sub-system according to claim 3, wherein said controller operates under program control for acquiring said sensor signals and generating control signals for application to said functional devices of said cogeneration system.~~

~~10. The control sub-system according to claim 9, wherein said controller operates according to said program control and at least acquires and takes into account an outdoor ambient temperature reading from one of said sensors before generating said control signals.~~

~~11. The control sub-system according to claim 10, wherein said controller determines a set-point for a hydronic supply temperature in the hydronic heating system from said outdoor ambient temperature reading.~~

~~12. The control sub-system according to claim 11, wherein said controller establishes said set-point according to a linear scale from 25°C at an outdoor temperature of 20°C to 75°C at an outdoor temperature of -20°C.~~

~~13. The control sub-system according to claim 11, wherein the heat source comprises a gas valve and a burner, and wherein said controller uses said set-point to operate the heat source in a variable capacity mode by modulating the gas valve on the burner to maintain actual hydronic supply temperature as sensed by one of said sensors at said set-point.~~

~~14. The control sub-system according to claim 13, wherein said controller further prevents working fluid in the liquid state from entering the expander by coordinating a fuel flow rate (heat input rate) to the burner with a feed flow rate of the working fluid exiting the feed pump.~~

~~15. The control sub-system according to claim 14, wherein the functional devices further include an evaporator fluidly connected to said expander by said piping and heated by said heat source, and said controller controls said fuel flow rate and said feed flow rate to maintain an exit temperature of the evaporator at 310°F (154.4°C).~~

~~16. The control sub-system according to claim 1, wherein the functional devices further include a hydronic pump circulating fluid in the hydronic heating system to and from heat exchange piping of the condenser, and wherein said controller controls the hydronic pump at a speed that maintains a pressure difference between the supply and return of the fluid to the heat exchanger piping of the condenser at a preselected value for optimal thermodynamic performance of the hydronic heating system.~~

~~17. The control sub-system according to claim 1, wherein the functional devices further include an evaporator fluidly connected between the feed pump and expander, and a desuperheater fluidly connected between the expander and condenser, said desuperheater having a return to the evaporator, and a switching valve provided in the piping for directing the organic working fluid either to the desuperheater or the evaporator directly, and wherein the controller controls the switching of the switching valve.~~

~~18. The control sub-system according to claim 1, wherein the expander is selected from a positive displacement expander and a scroll expander.~~

~~19. The control sub-system according to claim 1, wherein the functional devices further include a bypass valve and a shutoff valve in the piping to bypass and shutoff said superheat working fluid vapors from entering into the expander, the piping including a bypass loop connected between the bypass valve and condenser, and wherein said controller controls the opening/closing of the bypass valve and shutoff valve at least in response to startup and shutdown conditions of the cogeneration system.~~

~~20. The control sub-system according to claim 19, wherein said shutdown conditions include a heat call satisfied signal, a startup sequence failure, or exceeding a related preset value for an expander inlet temperature, an expander inlet pressure, a feed pump inlet temperature, a feed pump inlet pressure, a protection relay trip, or a power module temperature.~~

~~21. A control subsystem for governing the operation of a cogeneration system configured to operate with an organic working fluid, said system having a plurality of functional devices at least including a heat source, an expander having operatively coupled a generator to produce electricity, a condenser in fluid communication with said expander, and a pump configured to circulate said organic working fluid from said condenser through piping in thermal communication with said heat source such that heat transferred therefrom superheats said organic~~

~~working fluid to provide superheated organic working fluid vapors to said expander, said control subsystem comprising:~~

~~a plurality of sensors each providing a sensor signal indicative of a parameter of the cogeneration system;~~

~~a plurality of control points; and~~

~~a programmable controller coupled to said plurality of sensors and to said control points of said cogeneration system associated with functional devices thereof, said controller responding to said sensor signals to control said functional devices of the cogeneration system to thereby vary the operating characteristics of the cogeneration system.~~

~~22. A control system in combination with a cogeneration system having a plurality of functional devices and using an organic working fluid to heat a hydronic heating system and produce electrical power, the combination comprising:~~

~~a plurality of sensors for providing electrical sensor signals indicative of operating parameters of the cogeneration system;~~

~~a plurality of control points associated with said functional devices to change the operating parameters of the cogeneration system; and~~

~~a programmable controller coupled to the electrical sensors and to said control points of the system associated with said functional devices, said controller responding to the sensor signals and generating a plurality of control signals to control the operation of the functional devices; said controller defining a plurality of interactive control loops each generating a control signal for controlling a different one of said functional devices as a function of variation in a sensor signal supplied to the controller relative to at least a set point value for a control loop for the hydronic heating system, wherein the set point is determined by said controller from receiving an outdoor ambient temperature sensor signal from one of said sensors.~~

~~23. A method of controlling the thermal and electrical output of integrated micro combined heat and power generation systems used to supply domestic electrical power, domestic space heating (SH) water, and/or domestic hot water (DHW), and which converts heat energy~~

~~contained in superheated vapors of an organic working fluid to mechanical energy, and distributes the superheated vapors under pressure to at least one functional device having a heating need which varies over time, comprising:~~

~~monitoring over a period of time an ambient outdoor temperature to determine said heating need by said at least one functional device; and~~

~~changing in response to said ambient outdoor temperature indicating at any given time that a different amount of said superheated vapors than that being delivered to said at least one functional device is so needed to satisfy said heating need.~~

~~24. The method according to claim 23, wherein there are a plurality of process control sensors communicating with a controller programmed to change operating parameters of said system in response to said ambient outdoor temperature in order to furnish said at least one functional device a supply of said superheated vapors.~~

~~25. The method according to claim 24, wherein said ambient outdoor temperature is used to determine a desired hydronic supply temperature for a hydronic fluid and said amount of superheat vapors is varied to heat said hydronic fluid to said desired hydronic supply temperature.~~

~~26. The method according to claim 25, further comprising sensing an actual hydronic supply temperature via at least one of said process control sensors, and adjusting a firing rate of a burner used to heat said organic working fluid in order to bring said actual hydronic supply temperature into line with said desired hydronic supply temperature.~~

~~27. The method according to claim 26, further comprising increasing said firing rate if said actual hydronic supply temperature is too low, and decreasing said firing rate if said actual hydronic supply temperature is too high.~~

~~28. The method according to claim 26, further comprising sensing via at least one of said process control sensors temperature of said superheated vapors and adjusting flow rate of said working fluid past said burner to maintain said temperature of said superheated vapors within a desired operating parameter.~~

~~29. The method according to claim 28, wherein said desired operating parameter for said temperature of said superheated vapors is 310ºF (154.4ºC).~~

~~30. The method according to claim 28, wherein said flow rate is increased if said temperature of said superheated vapors is above said desire operating parameter, and decreased if said temperature of said superheated vapors is below said desired operating parameter.~~

1. An indirectly-heated micro combined heat and power system comprising:

a heat source;

an interloop heat exchanger in thermal communication with said heat source;

a first fluid-circulating loop with at least a portion thereof passing through a first channel of said interloop heat exchanger, said first fluid-circulating loop comprising:

an organic working fluid;

a scroll expander;

a generator operatively responsive to said scroll expander to generate electricity;

a condenser in fluid communication with said scroll expander, said condenser adapted to establish a heat exchange relationship between said organic working fluid and an external heat exchange fluid for space heating within a dwelling; and

a pump for the circulation of said organic working fluid; and

a second fluid circulating loop with at least a portion thereof passing through a second channel of said interloop heat exchanger such that said second fluid circulating loop is in thermal communication with said first loop, said second fluid circulating loop comprising:

a first sub-loop comprising:

piping to circulate a heat exchange fluid disposed in said second fluid-circulating loop, at least a portion of said piping in thermal communication with said heat source;

a domestic hot water heat exchanger; and

at least one pump to circulate a portion of said heat exchange fluid through said domestic hot water heat exchanger;

a second sub-loop comprising:

piping to circulate said heat exchange fluid such that it is in heat exchange relationship with said organic working fluid in said interloop heat exchanger;

at least one pump to circulate a portion of said heat exchange fluid through said interloop heat exchanger,

wherein said heat source, said heat exchanger, said first loop and said scroll expander are configured such that, upon application of heat from said heat source to said organic working fluid via said interloop heat exchanger, said organic working fluid becomes superheated to an extent that said organic working fluid remains superheated at least through said scroll expander.

2. An indirectly-heated micro combined heat and power system according to claim 1, further comprising an exhaust duct in fluid communication with said heat source such that products from said heat source may be removed from said micro combined heat and power system.

3. An indirectly-heated micro combined heat and power system according to claim 2, further comprising a heat exchanger in thermal communication with said exhaust duct.

4. An indirectly-heated micro combined heat and power system according to claim 1, further comprising a space heating loop preheat device placed in heat exchange communication with said second fluid circulating loop.

5. An indirectly-fired cogeneration system comprising:

a heat source;

a passive heat transfer element in thermal communication with said heat source;

a first circuit disposed adjacent an end of said passive heat transfer element such to accept heat transferred therefrom, said first circuit comprising:

an organic working fluid that becomes superheated upon receipt of heat from said passive heat transfer element;

a scroll expander configured to receive said superheated organic working fluid;

a condenser in fluid communication with said scroll expander, said condenser configured to transfer at least a portion of the excess heat contained in said organic working fluid to an external heating loop; and

a pump configured to circulate said organic working fluid through said first circuit;

a generator coupled to said scroll expander to produce electricity in response to motion imparted to it from said scroll expander; and

a second circuit configured to transport a heat exchange fluid therethrough, said second circuit in thermal communication with an end of said passive heat transfer element such that heat transferred therefrom increases the energy content of said heat exchange fluid, said second circuit comprising:

a combustion chamber disposed adjacent said heat source;

at least one external loop heat exchanger; and

conduit to transport said heat exchange fluid between said combustion chamber and said at least one external loop heat exchanger.

6. An indirectly-fired cogeneration system according to claim 5, wherein said passive heat transfer element is a heat pipe.

7. An indirectly-fired cogeneration system according to claim 5, wherein said combustion chamber is defined by:

an exhaust duct in combustion communication with said heat source;

an exhaust fan coupled to said exhaust duct to facilitate the removal of exhaust gas; and

an exhaust gas recirculation duct in exhaust communication with said combustion chamber.

8. A cogeneration system comprising:

a heat source;

a passive heat transfer element in thermal communication with said heat source;

a first circuit disposed adjacent an end of said passive heat transfer element such to accept heat transferred therefrom, said first circuit comprising:

an organic working fluid that becomes superheated upon receipt of heat from said passive heat transfer element;

a scroll expander configured to receive said superheated organic working fluid;

a condenser in fluid communication with said scroll expander, said condenser configured to transfer at least a portion of the excess heat contained in said organic working fluid to an external heating loop; and

a pump configured to circulate said organic working fluid through said first circuit; and

a generator coupled to said scroll expander to produce electricity in response to motion imparted to it from said scroll expander.

9. A cogeneration system according to claim 8, wherein said passive heat transfer element is a heat pipe.

10. An cogeneration system according to claim 8, wherein said combustion chamber is defined by:

an exhaust duct in combustion communication with said heat source;

an exhaust fan coupled to said exhaust duct to facilitate the removal of exhaust gas; and

an exhaust gas recirculation duct in exhaust communication with said combustion chamber.

ABSTRACT OF THE DISCLOSURE

~~Microprocessor-based control sub-systems which control the thermal and electrical output of integrated micro-combined heat and power generation (M-CHP) systems used to supply domestic electrical power, domestic space heating (SH) water, and domestic hot water (DHW). The M-CHP system uses a microprocessor controller to control the internal operating conditions, such as pump speeds, gas flow rate, and evaporator outlet temperature. Controlling these parameters enables setting the capacity of the system at any instant in time, thereby permitting load following, using a variable capacity operation. The controller also monitors through sensors a number of additional safety controls and system protection devices, such as relays/contactor of the alternator to grid, and electrical trips to the feed pump, the oil pump, the hydronic pump, the blower, the gas valve, the expander bypass valves, and other electrically powered devices in the system.~~

An integrated system to provide both heat and electric power. The integrated, or cogeneration, system operates with an organic working fluid that circulates in a Rankine-type cycle, where the organic working fluid is superheated by a heat source, expanded through an involute spiral wrap (scroll) expander such that the organic working fluid remains superheated through the expander, cooled in a condenser, and pressurized by a pump. Heat exchange loops within the system define hot water production capability for use in space heating and domestic hot water, while the generator is coupled to the scroll expander to generate electricity.